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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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JANUARY 1909

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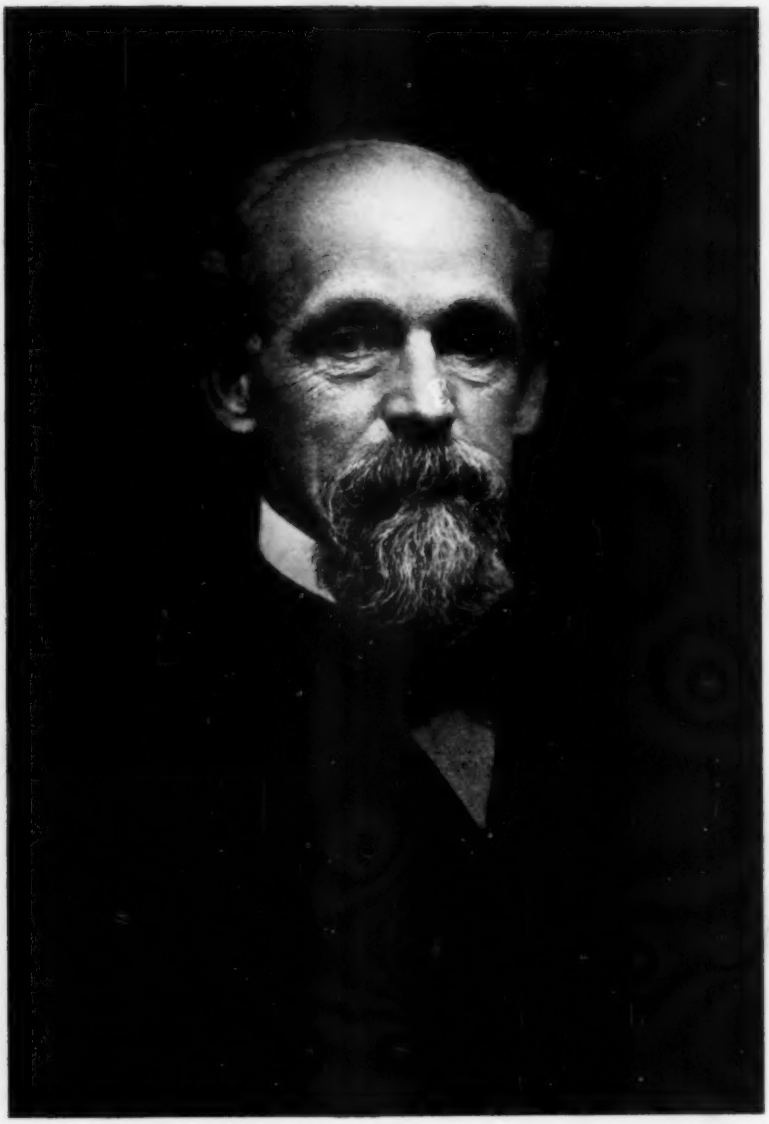

John R. Freeman

HONORARY MEMBER
OF
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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The Society as a body is not responsible for the statements of facts or opinions advanced in papers or discussions. C55.

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THE CONSERVATION IDEA AS APPLIED TO THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

PRESIDENTIAL ADDRESS 1908

By President M. L. Holman, St. Louis, Mo.

The rate at which the natural resources of the United States are being consumed caused the President to call a conference of the Governors of the States in 1908 to consider the questions of the conservation and use of the great fundamental sources of wealth of the Nation. That the question is considered a vital one, is shown by the fact that the conference marks the first time in the history of our country when the Governors of the States have assembled at the White House to consult with the Chief Executive of the Nation.

2 Invitations to participate in the Conference were issued to some 71 "National Organizations concerned in the development and use of" natural resources, and "to the Senators and Representatives in Congress; the Supreme Court; the Cabinet; and the Inland Waterways Commission." With the Governors were three men from each State chosen as advisers.

3 The Conference assembled in the East Room of the White House, Wednesday, May 13, 1908, and the morning session was taken up by the address of the President, on Conservation as a National Duty. During the sessions of Wednesday and Thursday the following papers were presented:

The Conservation of Ores and Related Minerals, by Andrew Carnegie.

The Waste of Our Fuel Resources, by Dr. White.

The Natural Wealth of the Land and its Conservation, by James J. Hill.

Soil Wastage, by Prof. T. C. Chamberlain.

Forest Conservation, by R. A. Long.

Resources Related to Irrigation, by George C. Pardee.

Grazing and Stock Raising, by H. A. Jastro.

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers. All papers are subject to revision.

4 A limited general discussion was had on each of these papers. The immediate result of the Conference could be an expression of opinion only, in the form of resolutions. Pursuant to a motion the president appointed a Committee on Resolutions, consisting of the Governors of South Carolina, Utah, Wisconsin, Louisiana and New Jersey.

5 The four national Engineering Societies were represented by their presidents, who formed a part of the audience and who jointly presented resolutions which went, with others, to the Committee on Resolutions. The object sought by these representatives was a broad treatment of the subject on non-political lines, and an avoidance of any indication of departmental jealousy. During the discussion it became apparent that some effort would be required to keep the Conference from political bias. On Friday morning the Committee on Resolutions being ready to report, the reading of the papers set for Friday was suspended and the resolutions prepared were introduced. The President presided and under his skilful parliamentary guidance the resolutions were adopted without a dissenting vote.

6 The report of the Committee has not been given the publicity which it deserves and I therefore give it in full, as published in the Cincinnati Inquirer of May 16, 1908:

We the Governors of the States and territories of the United States of America in conference assembled, do hereby declare the conviction that the great prosperity of our country rests upon the abundant resources of the land chosen by our forefathers for their homes and where they laid the foundation of this great nation.

We look upon these resources as a heritage to be made use of in establishing and promoting the comfort, prosperity and happiness of the American people, but not to be wasted, deteriorated or needlessly destroyed.

We agree that our country's future is involved in this; that the great natural resources supply the material basis upon which our civilization must continue to depend, and upon which the perpetuity of the nation rests.

We agree, in the light of facts brought to our knowledge and from the information received from sources which we cannot doubt, that this material basis is threatened with exhaustion. Even as each succeeding generation from the birth of the nation has performed its part in promoting the progress and development of the republic, so do we in this generation recognize it as a high duty to perform our part, and this duty in large degree is in the adoption of measures for the conservation of the natural wealth of the country.

We declare our firm conviction that this conservation of our natural resources is a subject of transcendant importance, which should engage unremittingly the attention of the nation, the State and the people in earnest coöperation. These natural resources include the land on which we live and which yields our food; the living waters which fertilize the soil, supply power and form great avenues of commerce; the forests which yield the materials for our homes, prevent erosion

of the soil, and conserve the navigation and other uses of our streams; and the minerals, which form the basis of our industrial life, and supply us with heat, light and power.

We agree that the land should be so used that erosion and soil wash should cease; that there should be reclamation of arid and semi-arid regions by means of irrigation, and of swamp and overflowed regions by means of drainage; that the waters should be so conserved and used as to promote navigation, to enable the arid regions to be reclaimed by irrigation and to develop power in the interests of the people; that the forests, which regulate our rivers, support our industries and promote the fertility and productiveness of the soil, should be preserved and perpetuated; that the minerals found so abundantly beneath the surface should be so used as to prolong their utility; that the beauty, healthfulness and habitability of our country should be preserved and increased; that the source of national wealth exists for the benefit of all the people, and that the monopoly thereof should not be tolerated.

We commend the wisdom of the President in sounding the note of warning as to the waste and exhaustion of the national resources of the country, and signify our appreciation of his action in calling this Conference to consider the same, and to seek the remedies therefor through coöperation of the nation and the states.

We agree that this coöperation should find expression in suitable action by the Congress within the limits of and co-extensive with the national jurisdiction of the subject and complementary thereto by the Legislatures of the several States within the limits co-extensive with their jurisdiction.

We declare in convention that in the use of the national resources our independent states are interdependent and bound together by ties of mutual benefits responsibilities and duties.

We agree in the wisdom of future conferences between the President, members of Congress and the Governors of the States, regarding the conservation of our natural resources with the view of continued operation and action on the line suggested, and to this end we advise that from time to time, as in his judgment may seem wise, the President call the Governors of the States, members of Congress and others into conference.

We agree that further action is advisable to ascertain the present condition of our national resources and to promote the conservation of the same. And to that end we recommend an appointment by each State of a commission on the conservation of natural resources, to coöperate with each other and with any similar committee on behalf of the Federal Government.

We urge that continuation and extension of forest policies be adopted to secure the husbanding and renewal of our diminishing timber supply, prevention of soil erosion and the protection of head waters and the maintenance of the purity and navigability of our streams. We recognize that the private ownership of forest land entails responsibilities in the interests of all the people, and we favor the enactment of laws looking to the protection and replacement of privately owned forests.

We recognize in our waters a most valuable asset of the people of the United States, and we recommend the enactment of laws looking to their conservation, to the end that navigable and source streams may be brought under complete control and fully utilized for every purpose. We especially urge on the Federal Congress the immediate adoption of a wise, active and thorough waterway policy

providing for the prompt improvement of our streams and conservation of their water-sheds required for the uses of commerce and the protection of the interests of our people.

We recommend the enactment of laws looking to the prevention of waste in the mining and extraction of coal, oil, gas and other minerals with a view to their wise conservation for the use of the people and to the protection of human life in the mines.

Let us conserve the foundations of our prosperity.

FUNDAMENTAL CONSTANTS

7 As it is the peculiar function of the engineer to assist in making some of the resources of nature available for the use and convenience of man, we may well spend our time this evening considering a few of the phases of the problem. The President particularly desired the coöperation in the movement of the engineers of the United States and subsequently ascribed to the action of the Engineering Societies the credit of inaugurating the conservation campaign on non-political lines.

8 As it was the rapidly increasing rate of consumption of our natural resources that induced the President to make a new departure along the line of "States' Rights" and to call on the Governors for a conference, let us first examine into the forces at work consuming our national inheritance and turn our attention to ourselves, as the primary cause or force that is so rapidly dissipating the natural resources of the nation and emptying the store house of nature. For purposes of the present discussion we will confine ourselves to the territory of the United States. Table 1, compiled from the census returns, gives in a condensed form the fundamental constants and the independent variables of our problem, that is, our lands and inland waters and our population, as they stood in 1900.

TABLE 1 LAND AND WATER, AND POPULATION OF THE STATES OF THE UNITED STATES

State or Territory	SQUARE MILES			Population	Density
	Gross Area	Water	Land		
Alabama	52,250	710	51,540	1,828,697	35.5
Arizona	113,020	100	112,920	122,931	1.1
Arkansas	53,850	805	53,045	1,311,564	24.7
California	158,360	2380	155,980	1,485,053	9.5
Colorado	103,925	280	103,645	539,700	5.2
Connecticut	4,990	145	4,845	980,420	187.5
Delaware	2,050	90	1,960	184,735	94.3

TABLE 1—Continued

State or Territory	SQUARE MILES			Population	Density
	Gross Area	Water	Land		
District of Columbia	70	10	60	278,718	4645.3N.B.
Florida	58,680	4440	54,240	528,542	9.7
Georgia	59,475	495	58,980	2,216,331	37.6
					1.9
Idaho	84,800	510	84,290	161,772	
Illinois	56,650	650	56 000	4,821,550	86.1
Indiana	36,350	440	35,910	2,516 462	70.1
Indian Territory	31,400	400	31,000	392,060	12.6
Iowa	56,025	550	55,475	2,231,853	40.2
Kansas	82,080	380	81,700	1,470,495	18.0
Kentucky	40,400	400	40 000	2,147,174	53.7
Louisiana	48,720	3300	45,420	1,381,625	30.4
Maine	33,040	3145	29,895	694,466	23.2
Maryland	12,210	2350	9,860	1,188,044	120.5
Massachusetts	8,315	275	9,040	2,805,346	348.9
Michigan	58,915	1485	57,430	2,420,982	42.2
Minnesota	83,365	4160	79,205	1,751,394	22.1
Mississippi	46,810	470	46,340	1,551,270	35.5
Missouri	69,415	680	68,735	3,106,665	45.2
Montana	146,080	770	145,310	243,329	1.7
Nebraska	77,510	670	76,840	1,066,300	13.9
Nevada	110,700	960	109,740	42,335	0.4
New Hampshire	9,305	300	9,005	411,588	45.7
New Jersey	7,815	290	7,525	1,883,669	250.3
New Mexico	122,580	120	122,460	195,310	1.6
New York	49,170	1550	47,620	7,268,894	152.6
North Carolina	52,250	3670	48,580	1,893,810	39.0
North Dakota	70,795	600	70,195	319,146	4.5
Ohio	41,060	300	40,760	4,157,545	102.0
Oklahoma	39,030	200	38,830	398,331	10.3
Oregon	96,030	1470	94,560	413,536	4.4
Pennsylvania	45,215	230	44,985	6,302,115	140.1
Rhode Island	1,250	197	1,053	428,556	407.0
South Carolina	30,570	400	30,170	1,340,316	44.4
South Dakota	77,650	800	76,850	401,570	5.2
Tennessee	42,050	300	41,750	2,020,616	48.4
Texas	265,780	3490	262,290	3,408,710	11.6
Utah	84,970	2780	82,190	276,749	3.4
Vermont	9,565	430	9,135	343,641	37.6
Virginia	42,450	2325	40,125	1,854,184	46.2
Washington	69,180	2300	66,880	518,103	7.7
West Virginia	24,780	135	24,645	958,800	38.9
Wisconsin	56,040	1590	54,450	2,069,042	38.0
Wyoming	97,890	315	97,575	95,531	0.9
Totals and Averages			2,971,038	76,357,575	25.7

TABLE 2 PUBLIC LANDS, JULY 1, 1908

FROM REPORT OF THE GENERAL LAND OFFICE OF THE UNITED STATES

States	Acres	States	Acres
Alabama	129,713	Montana	46,532,440
Alaska	368,021,509	Nebraska	3,074,658
Arizona	42,769,202	Nevada	61,177,050
Arkansas	1,060,185	New Mexico	44,777,505
California	29,872,493	North Dakota	2,322,150
Colorado	23,696,697	Oklahoma	86,339
Florida	414,942	Oregon	16,957,913
Idaho	26,785,002	South Dakota	6,561,295
Kansas	171,446	Utah	36,578,998
Louisiana	116,249	Washington	4,635,001
Michigan	135,551	Wyoming	37,145,302
Minnesota	1,788,705		
Mississippi	42,791	Total	754,895,296
Missouri	27,480		

9 While dealing with statistics of population, it may not be amiss to call attention to the fact that we are not always governed by the will of the majority. Table 3, from the census returns, shows quite the opposite; in fact it is about an even chance that a successful presidential candidate represents the majority of the voting population:

TABLE 3 PRESIDENTIAL VOTE

Year	Total popular vote	WINNING CANDIDATE		
		Popular vote	Per cent of total	Per cent of population
1828	1,156,328	647,231	55.97	....
1832	1,250,799	687,502	54.96	....
1836	1,498,205	761,549	50.83	....
1840	2,410,778	1,275,017	52.89	14.1
1844	2,698,611	1,337,243	49.55	....
1848	2,861,908	1,360,101	47.36	....
1852	3,138,301	1,601,474	50.90	....
1856	4,053,967	1,838,169	45.34	....
1860	4,676,853	1,866,352	39.91	15.2
1864	4,024,792	2,216,067	55.06	....
1868	5,724,684	3,115,071	52.67	....
1872	6,466,165	3,597,070	55.63	....
1876	8,412,733	4,033,950	47.95	....
1880	9,209,588	4,449,053	48.26	18.3
1884	10,044,985	4,911,017	48.48	....
1888	11,372,299	5,440,216	47.83	....
1982	12,059,351	5,556,918	45.73	....
1896	13,923,202	7,104,779	50.49	....
1900	13,967,974	7,217,810	51.66	18.4
1904	13,513,995	7,620,670	57.13	....

10 It will be noted that President Lincoln was elected by 40 per cent of the popular vote and that President Roosevelt received the greatest per cent of the popular vote since 1828. As new parties are developed the chances of government by the minority become greater, and with a sufficient number of political parties in the field, revolutions will be the order of the day.

HEALTH

11 Of our many resources, the most important one is the health of the people. The safeguarding of the general health and the prevention of preventable diseases form a large field for the practical operation of the conservation idea and will require the best efforts of the State and the Nation, the sanitary engineer and the health officer. The experience of the last decade has shown that prompt action by competent specialists backed by the power of the Government will conserve to the State, the community and the citizen, an earning power and working capacity that would otherwise be lost or diminished. The preventive work, particularly in zymotic diseases, must fall to the sanitarian and not to the physician, unless we adopt the Chinese practice of paying our doctors to keep us in good health and stop the honorarium when we fall sick.

12 A sharp distinction must be drawn between protecting people from unnecessary sickness and restoring the sick to health. The medical profession is an ancient and honorable one. It has accomplished a vast amount of good and has furnished a multitude of noble examples of devotion to duty. It is not my purpose to criticise the profession, but to draw attention to the fact that we need and must have comprehensive organizations, State, National, and municipal, which have for their field of work the protection of health. The training of the physician, and particularly his experience, are directed along lines that develop the specialist, but as a general rule executive training and experience are lacking. Our municipal records furnish much evidence to the fact that successful physicians, called from private practice, are totally devoid of executive ability. There have been exceptions, but they only emphasize the rule. Some States and cities now have Boards of Health, but for the most part they are composed of physicians and the work handled is the cure and not the prevention of disease. Less than one-half of the States record all deaths and require burial permits, and but few States keep accurate accounts of epidemics and contagious diseases.

13 Our school children are not properly protected from zymotic diseases. Some work in this direction, however, is now being undertaken by New York City. Our business interests suffer large losses from epidemics and generally work to prevent quarantine and other evidences of the true state of affairs from becoming public. Too much emphasis cannot be put on this part of the conservation problem. The public health is our most valuable resource and its protection and conservation is of the utmost importance.

14 One of the first problems in the field of sanitation is the protection of our rivers, lakes and other sources of domestic water supply from pollution. At present most, if not all, of our inland cities and towns discharge raw sewage into the natural water courses and lakes. One notable city for years discharged its sewage into the margin of a lake and took its water supply from the same place. Typhoid fever, resulting from this arrangement of sewage disposal and water supply, was prevalent and widespread. The general Government, through a National Sanitary Commission should have intervened for the protection of the citizens of other states and the residents of the city. This condition, in a slightly modified degree, still exists, and the necessity for a sanitary overhauling is evident to any unbiased student. Of late a portion of the sewage has been diverted out of the natural water-shed of the lake, but the conditions are far from safe, and a movement is now on foot to divert the remaining sewage ultimately, under the guise of an improved water-way. In this particular instance the sewage should be purified and discharged at the cost of the municipality producing it. An effort to force this treatment of the case resulted in a "Scotch verdict" and the present conditions may be expected to continue until there is a general movement looking to the protection of our water courses from pollution, or until each community develops a distinct bacillus that can be positively identified. In the matter of protecting our rivers and water courses from pollution, we are far behind the times and should begin our conservation era with laws prohibiting all parties, private and public, from burdening the country with filth that should be handled by the parties producing it, and at said parties' own proper cost and expense.

15 In addition to the protection of our sources of water supply the purification of the water furnished to cities must be given attention. It is an open question to what extent the State should exercise its power in this direction, but there is no question as to the results which an impure water supply entails, and the responsibility, morally if not legally, of the municipality which furnishes polluted water. It is my

opinion that the State should require the municipality, which it empowers to levy taxes, to discharge its full duty to its citizens. When a private water company supplies water it is under obligation to furnish a safe drinking water, just as fully as it would be to furnish adequate fire protection.

16 We have not the time this evening to enter into a detailed discussion of any of the problems involved in water purification, but an examination of the results obtained in the city of Washington for the years 1896 to 1897 is ample evidence to convince anyone of what may be accomplished. Numerous other cases might be cited showing good returns in life and health. The evidence is convincing, and the conclusion certain that the residents of our country suffer much bodily harm and unnecessary loss of life from drinking diluted sewage. Life insurance companies make us pay high rates for our negligence and pile up vast sums of money which is often manipulated to increase our burdens further. The reports of the insurance commissioner of the State of Connecticut show for 19 of the large insurance companies total assets amounting to the enormous sum of $2,718,631,737 and liabilities of $2,591,534,168.

FOOD

17 The next phase of the problem relates to the individual, and is not primarily within the field of the engineer, nevertheless the mechanical engineer has been at the bottom of a share of the trouble, by way of the cold-storage warehouse. No fault is to be found with the work of the engineer, but he has put into the hands of the people a process that is often made use of to foist on the public edibles not fit for human consumption. The chemist and the chemical engineer have discovered and manufactured anti-ferments until it has become necessary for the Government to ascertain by experiment how much chemical preservative the ordinary citizen could consume, and to require the labeling of manufactured foods. However, the labeling of our catsup bottles with the legend that one-tenth of one per cent of benzoate of soda is contained therein does not seem to deter its use. The rate of abuse now prevalent in some sections in storing undrawn poultry, and in exposing chilled meats in transit from storage to retail butcher shops will soon require the Government to regulate the cold-storage warehouse business. It is not apparent which resources the pure food law was intended to conserve. With the paternal form of government obtaining in some of the European

countries, the health of the individual is the paramount resource. With us, however, "civil and religious liberty" seems to include unnecessary exposure to disease. At the Conference in Washington the preventable disease problem was practically overlooked, perhaps from the fact that no trust seems to be operating in that field.

LAND

18 It is a question which is our chief natural resource, ourselves or our country. I have assumed the citizen as the first, and now come to the second, the land which we inhabit. In defence of this classification it is sufficient to mention the fact that the present development of the United States can be traced to small beginnings on our Eastern coast a few years ago, and while not within the province of the engineer to discant on the "rise and fall of nations," it may not be amiss to call attention to the necessity of keeping alive the sterling qualities and sound principles of the pioneers as a safeguard to our future progress as a republic. Our land may well receive our careful attention and its conservation be thoroughly studied. We shall be face to face in the near future with the fact that there is not sufficient land to support us at our present rate of growth and methods of living.

19 We will leave the Malthusian end of this problem to the political scientist and philosopher and direct our attention to the engineering side of the case. The accompanying tables show some of the more important items for the States and Territories. Foreign possessions

TABLE 4 INCREASE OF POPULATION

Year of census	Total population	Immigration	Per cent increase of population	Per cent of immigration	Foreign born	Per cent foreign born
1790	3,929,214	.....	.....	....	.....	....
1800	5,308,483	.....	35.1	....	.....	....
1810	7,239,881	.....	36.4	....	.....	....
1820	9,638,453	.....	33.1	....	.....	....
1830	12,866,020	.....	33.5	....	.....	....
1840	17,069,453	.....	32.7	....	.....	....
1850	23,191,876	.....	35.9	....	2,244,602	9.7
1860	31,443,321	2,598,214	35.6	8.3	4,138,697	13.2
1870	38,558,371	2,314,824	22.6	6.0	5,567,229	14.4
1880	50,155,783	2,812,191	30.1	5.6	6,679,943	13.3
1890	62,622,250	5,246,613	24.9	8.4	9,308,104	14.8
1900	75,568,686	3,687,564	20.7	4.8	10,460,085	13.7

TABLE 5 COMPARISON OF URBAN POPULATION AND AVAILABLE FARM LAND

Year of census	Urban population	Per cent of urban	Number of farms	Total acres	Average acres per farm	Per cent increase of farms
1790	131,472	3.4	.....	.....	.....	....
1800	210,873	4.0	.....	.....	.....	....
1810	356,920	4.9	.....	.....	.....	....
1820	475,135	4.9	.....	.....	.....	....
1830	864,509	6.7	.....	.....	.....	....
1840	1,453,994	8.5	.....	.....	.....	....
1850	2,897,586	12.5	1,449,073	293,560,614	202.6	....
1860	5,072,256	16.1	2,044,077	407,212,538	199.2	41.1
1870	8,071,875	20.9	2,659,985	407,735,041	153.3	30.1
1880	11,318,875	22.6	4,008,907	536,081,835	133.7	50.7
1890	18,272,503	29.2	4,564,641	623,218,619	136.5	13.9
1900	24,992,199	33.1	5,739,657	841,201,546	146.6	25.7

In the year 1900, on the basis of 4000 and over, the percentage of urban population was 37.3.

are not included, as they do not enter into the Problem except as disturbing elements, and it is to be hoped that the franchise will not be extended to them. We must fight out our political differences at home and not wait for returns from the Philippines to know who is to be our next President.

20 Texas became one of the United States in 1845. If we assume the growth in population of the States and territories from the year 1850 to the year 1900 as an indication of the rate of growth which we may expect, that is, judge the future by the past, we can develop

TABLE 6 AGRICULTURAL POPULATION

Year of census	Population engaged in agricultural pursuits	Per cent of total population	Average value of farms per acre	Per cent increase in value
1880	7,713,875	44.3	$22.72	36.2
1890	8,565,926	37.7	25.81	32.0
1900	10,381,765	35.7	24.39	27.6

a formula that fairly represents the case. Representing the population at any time, T years subsequent to 1850, by P, we find that the expression:

$$\text{Log } P = 1.702 \, (\log T - 0.897)$$

gives the approximate results shown in Table 7.

21 We see that the formula gives results corresponding with the census for the years 1850, 1890, and 1900. The computed results

TABLE 7 COMPARISON OF COMPUTED AND ACTUAL RATES OF GROWTH

Year	Population as shown by census, millions	Population as given by formula, millions	Error of formula, millions excess
1850	23	23	1
1860	31	32	2
1870	39	41	2
1880	50	52	....
1890	63	63	....
1900	75	75	....

for 1860, 1870, and 1880 are high, but the effect of the Civil War may be assumed to be responsible for a large part of the loss. Now *if* this formula holds good, as a rough approximation, for the next 150 years we may expect the growth shown in Table 8.

TABLE 8 ESTIMATED FUTURE POPULATION OF THE UNITED STATES

Year	Estimated population, millions	Estimated density per square mile	Year	Estimated population, millions	Estimated density per square mile
1910	89	30	1960	168	57
1920	103	35	1970	186	63
1930	118	40	1980	202	68
1940	134	45	1990	225	76
1950	150	51	2000	245	83

22 The formula is, in my opinion, conservative and gives results that will probably be exceeded up to 1950. Beyond that time the reverse may be true. In round numbers, we may expect a population of 150 000 000 by the year 1950 and at the end of the present century we will be near the 250 000 000 mark. Our development will be both agricultural and industrial, and our limiting density of population will be governed by the ability of succeeding generations to compete in the markets of the world. A superficial study of the data will show that while our density will not become excessive during this century, we will nevertheless have to change our present extravagant pioneering methods of working and living.

23 Free trade would result in rapid changes in density of population, lowering the figures for the manufacturing states and raising them for the agricultural states. This problem is not for the Engineer, however, but he will form a part of the hegira while the statesmen are solving the tariff question.

TABLE 9 DENSITY OF POPULATION OF THE NEW ENGLAND STATES

Indicating the possibilities of an industrial development when the market for manufactured goods is protected from foreign competition

States	Total area Square miles	Water area Square miles	Land area Square miles	Population	Density per square mile
Maine	33,040	3,145	29,895	694,466	23.2
New Hampshire	9,305	300	9,005	411,588	45.7
Vermont	9,565	430	9,135	343,641	37.6
Massachusetts	8,315	275	9,040	2,805,346	348.9
Rhode Island	1,250	197	1,053	428,556	407.0
Connecticut	4,990	145	4,845	908,420	187.5
Totals and averages			62,973	5,592,017	174.98

TABLE 10 RATE OF GROWTH OF THE NEW ENGLAND STATES

State	1790	1800	1810	1820	1830	1840
Maine	3.2	5.1	7.7	10.0	13.4	16.8
New Hampshire	15.8	20.4	23.8	27.1	29.9	31.6
Vermont	9.4	16.9	22.9	25.8	30.7	32.0
Massachusetts	47.1	52.6	58.7	65.1	75.9	91.8
Connecticut	49.1	51.8	54.1	56.8	61.4	64.0
Rhode Island	63.4	63.7	70.9	76.6	89.6	100.3
Average	31.3	35.1	39.6	43.6	50.2	56.1
	1850	1860	1870	1880	1890	1900
Maine	19.5	21.0	21.0	21.7	22.1	23.2
New Hampshire	35.3	36.2	35.3	38.5	41.8	45.7
Vermont	34.4	34.5	36.2	36.4	36.4	37.6
Massachusetts	123.7	153.1	181.3	221.8	278.5	348.9
Connecticut	76.5	95.0	110.9	128.5	154.0	181.5
Rhode Island	136.0	160.9	200.3	254.9	318.4	407.0
Average	66.6	83.3	97.4	116.4	141.9	175.0

24 The figures are given to emphasize the fact that we have built up a condition in the United States that will require very careful handling to avoid disaster. The tariff has built up the trusts, and the trusts have built up the country, and the change from present conditions must be made slowly to avoid destruction.

25 The density of population of Germany for the year 1900 was 270 per square mile and it has been steadily increasing since that time. A comparison of this figure with the density of population of the

United States will convince the student that the German nation long ago solved most, if not all, of the conservation problems brought to our attention by the President. I venture to suggest that we might make progress by ascertaining the secret of German frugality and prosperity rather than by compiling masses of figures to prove that which is well known, viz: that we are wasting the resources of nature like a true prodigal son.

THE FOREST

26 One of the resources which the pioneer finds at hand and available for immediate use is the forest. It was necessary in many cases to clear away the primeval forests in order to render the land suitable for habitation, and this clearing had to be done at the minimum expenditure of labor. At present, however, the situation is very different. We are consuming over eight times as much lumber per capita as is used in Europe and our timber supply in the older parts of the country has been exhausted. It is high time we were taking thought for the future and making provision for a supply of timber for our successors. A good example of what may be accomplished in this line is to be found in Germany. Our own national Government has been making well directed efforts in this direction, so far as the public lands are concerned, but the states must each work in their respective domains in order to accomplish the best results. The clearing of land for farming is not criticised, but the unnecessary waste of timber resulting from an inordinate greed for gain (stimulated by a tariff) is the thing that needs checking. At this stage of the problem we begin to come into contact with the position of the engineer (I use the word in its broad sense) in relation to this natural resource. He has designed and built the machinery that has made the large production of lumber possible. The slow whip-saw and broadaxe methods of our forefathers have given way to the "shot gun feed," and any ordinary saw-mill will turn out more lumber in a day than the "Pilgrim Fathers" could have produced in a year.

27 The question that the engineer faces in private practice is how to increase the output and decrease the cost per thousand feet board measure. I have personally devised ways and means of *increasing* the fuel consumption of the power plant of a large saw mill, as a means of saving in the operating expenses of the mill. The problem, so far as the mill was concerned, was to produce lumber at the lowest price per thousand feet. A condition, and not a theory, confronted the Engineer. The alternative of meeting competition or

going out of business was the problem in this case as in many others, and this same test, under our system of government, is the force that will conserve those of our resources which fall in the class designated by the President as renewable.

28 Some of our manufacturing concerns using timber as a "raw material" are now, and have been for some time past, taking necessary precautions to insure a continuity of business existence. The preservation of forests for other purposes than as a source of supply for timber is "another story," and the questions involved are being worked out by the Bureau of Forestry, under the able direction of Gifford Pinchot. The reservation of forest lands by the Government needs careful handling to insure the protection of the citizen and to see that our timber supply is not captured *en masse* in the future by consolidated capital. A large surplus of capital or labor materials, generally makes trouble, and care must be taken to see that the national government does not lay foundations that may be used for a timber trust of gigantic proportions.

29 The waste of timber in the process of manufacture is at present as high as 75 per cent in some cases, and this waste is not counted from the log, but from the dimension lumber. Why is it that an English firm can sell shuttles and spindles in this country, in the face of a tariff? They do not waste anything, and the list of by-products made by them from scrap lumber would stagger a professor. A study of this problem for a New England concern, however, indicated that until the price of logs went up and the price of labor came down, by-products were not profitable undertakings in that locality.

30 The first action reserving public forest lands was due to the Act of Congress of 1891. Since that time the movement has progressed until we now have 150 National Reservations in the United States (See Table 11).

TABLE 11 FOREST RESERVATIONS

States	Acres	States	Acres
Arizona	9,422,125	Oklahoma	60,800
California	22,278,631	Oregon	16,463,535
Colorado	15,748,772	South Dakota	1,263,720
Idaho	20,336,427	Utah	7,127,339
Kansas	97,280	Washington	12,065,500
Montana	20,402,663	Wyoming	9,020,475
Nebraska	556,072		
Nevada	2,248,999	Total	142,972,855
New Mexico	8,279,584		

31 Our present supply of standing timber is estimated by Herbert M. Wilson at about 1 400 000 million ft. board measure; the annual "cut" at 100 000 million, a rate that will exhaust the supply in 14 years. We use per capita per year about 500 ft. board measure, as against an average of about 60 ft. for Europe. Our use of timber in house building not only consumes our supply but entails enormous expense for insurance and fire protection. Estimates of the fire tax for the year 1907 run about $2.50 per capita, as against an estimated average of $0.35 for Europe. The total cost for fire losses and for fire protection is not known, but the Government is at work on the problem and may be able to give some estimate in the next census report.

32 Our suggestion for reducing the consumption of timber is by the substitution of fibre products. This is now done to a large extent by the use of paper, and the use can be increased. In Germany peat products are used to replace wood where practical. The waste of fibre-producing material at present is enormous and the sources of future supply practically undiscovered.

WATER

33 Following the President's classification of our natural resources we turn to the one great natural phenomenon which makes our country suitable for the growth of a great nation and the development of a great people. The thing we all must have is water, and for that we are dependent on rainfall. Those of us who have not experienced the hardships of a scant and precarious water supply can have no appreciation of the bounties nature has lavishly scattered through this section of the country, nor of the local conditions created by deficient rainfall in the irrigation regions. In this as in other matters we acquire true knowledge only by hard knocks and bitter experience.

34 The Irrigation Engineer has to deal with these problems, and to aid in the conservation of the water resources of the country. The people of the irrigation districts, however, are the true promoters of the conservation idea as applied to water, and have made laws which have as their foundation the beneficial use of the water. Necessity has been the teacher and the people have learned their lessons well: as our resources become scarce necessity will teach us the same lesson, and eventually the doctrines of appropriation and application to beneficial use will become general, for the necessities.[1]

[1] For the luxuries our present methods will probably remain unchanged.

35 Our present laws and methods of irrigation are of comparatively recent date and are in the formative period, but the beginnings of the art antedate history. Irrigation was practiced by the peoples of ancient China and India and is mentioned in the early Biblical writings. It was evidently practiced by the inhabitants of the arid regions of this country also in prehistoric times. Life could not have been supported without it. Our general practice and methods were first derived from Mexico and show the results of Spanish law and practice. In the United States, irrigation was first practiced by the Mormons. In some parts of the country, irrigation followed mining, but the underlying principles governing the right to use water were fixed by pioneer necessities and followed Mexican practice.

36 The first effort of the Irrigation Engineer must be to divest himself of his unconscious prejudice in favor of the English law of riparian rights, which is set aside under the conditions which call the irrigation works into existence. The State claims the fee, and grants only an easement known as a water right. Such an easement may be acquired: *first*, by an appropriation, a notice of which is required to be posted in some states, while a mental process on the part of the appropriator seems sufficient in others; *second*, by the beneficial use of the water. The appropriator acquires the right to only as much water as he can use for beneficial (?) purposes and not the right to waste water. The process of law required by the State must be complied with and when a decree is issued as a result of the entire process the appropriator becomes possessed of an "adjudicated water right" which is to all intents and purposes property, but which may be abandoned and lost by non-use. Decreed water rights are classified as to priority, and the principle of "first in time is first in right" is strictly enforced. The appropriators of water on a stream are entitled to water in the order of their priorities, which bear the date of the appropriation, and in times of scarcity the last appropriator is the first to be shut down. This shutting down of headgates continues with a growing deficiency of water until in cases of extreme drought the only ditch operating will be the oldest priority. It is the operation of this principle that has led to the storing of water in times of plenty for use during the latter part of the season when water is scarce. The right to take water from the stream for direct application to land during the irrigating season is called a ditch right, the water being conducted from the stream by means of ditches and flumes. Where the water is first conducted into a reservoir either for use during the same season or to be held from one season to another, the right is called a

storage right. Some storage rights are for the non-irrigating or storage season, and others are for the storage of early floods for use later in the same season. In general, water rights are either ditch rights or storage rights. Water rights are classified in accordance with the importance of their use, viz:

Domestic rights, including municipal water supply.
Irrigation rights, ditch and storage.
Power rights.

37 These rights rank as to priority, independent of classification; that is, a power right of the lowest class may have the oldest priority and the first right to take water. A power right, however, can be condemned for a higher use, such as irrigation or domestic use, and in turn an irrigation right may be condemned for municipal use, but the rule does not work in the reverse order.

38 The amount of water taken by an appropriator is regulated and controlled by a head-gate built in the bank of the stream where the water is drawn out, and under the charge of State officials who set them to regulate the rate of flow from stream to ditch. In early days, ditch appropriations were made in inches of water, but the latter appropriations, and most of the priorities, are recorded in cubic feet of water per second of time. Some appropriations call for as much water as will flow through a ditch of given dimensions and grade. The inch of water, variously known as miner's inch, sluicing inch and irrigator's inch, is used to designate the rate at which water flows through the head-gate or measuring box. In various states and localities the "inch" represents different rates of flow. While it is an approximate measure, the inch is, nevertheless, a very convenient unit. Many outstanding contracts have the inch as the unit of measurement and decrees are occasionally issued in which the right to water is limited by the number of inches of water decreed. At present the tendency of the engineering profession is to use the "second-foot" and the "acre foot" as the units of rate and quantity. The inch is a handy unit for the irrigator and he will continue to use it, but the engineer must adapt himself to the location and ascertain the rate of flow represented by the local inch. The following memoranda on the inch of water in different states are given for the benefit of members who may need them at some future time:

a The Colorado Statute defines the inch as follows: "Every inch shall be considered equal to an inch-square orifice under a 5 in. pressure and a 5 in. pressure shall be from the top of the orifice of the box put into the bank of the ditch to the surface of the water."

b The Idaho Statute prescribes: "The amount of water that will flow through an orifice one inch square with a 4 in. pressure above the center of the orifice."

c The North Dakota Statute is: "The Miner's Inch shall be regarded as $\frac{1}{40}$ cu. ft. per second in all cases except when some other equivalent of the cubic foot per second has been stated by contract, or has been established by actual measurement or use."

d In Utah: "Standard measurement of flow, one cubic foot per second, known as the second-foot. Standard measurement of volume the acre-foot."

e The Wyoming Statute is: "A cubic foot per second of time shall be the legal standard for the measurement of water in this State, both for the purpose of determining the flow of water in natural streams and for the purpose of distributing the water thereof."

f Montana: A new act provides that "the cubic foot per second shall be the standard of measurement; 100 miner's inches, equivalent to $2\frac{1}{2}$ cu. ft. per second, not to affect measurements heretofore decreed by court."

g Nebraska: Every inch considered to be one inch square orifice under a 4 in. pressure, and a 4 in. pressure shall be from the top of the orifice in the measuring box to the surface of the water. "Boxes must be 6 in. in height, except where less than 12 in. are delivered; have descending slope $\frac{1}{8}$ in. to the foot and 14 ft. long; 50 miner's inches under 4 in. pressure equivalent to 1 cu. ft. per second of time.

h California: The following is from the San Francisco Post, of the late 70's: A miner's inch of water is a quantity that will flow through an inch aperture with a free discharge and under a constant pressure of 6 in. above the top of the opening. An aperture of 12.25 in. by 15.75 in., under a pressure of 6 in. above the top of the opening, will discharge 200 in., and is the basis of all measurements where water is retailed in small quantity in the States of California and Nevada."

39 It is customary to express the ratio between the inch and the second-foot as the number of inches which equal a second-foot. The values generally used are:

Colorado, miner's inch.
50 in. = 1 sec-ft; Statutory inch: 38.4
California, miner's inch:
50 in. = 1 sec-ft.
Arizona, miner's inch:
40 in. = 1 sec-ft.
New Mexico, miner's inch:
50 in. = 1 sec-ft.

The inch varies from 50 to the second foot to about 34 to the second foot and the engineer must ascertain the inch in use in the locality for which he is designing work.

40 In some states the duty of water is prescribed by law. By this is meant the duty which a given flow or volume is expected to perform and this is a measure of the quantity which the appropriator is permitted to take for his own use. In states with codes modeled after that of Wyoming, the rate of flow is fixed from 1 sec-ft. for 50 acres to 1 sec-ft for 70 acres. In Colorado the old standard seems to have been 1 in. per acre, and present practice tends to a flow of 1.44 sec. ft. for 80 acres.

41 Experience has demonstrated the potentiality of the soil of the arid regions, proving that with proper handling good crops can be raised and money made by the farmer. In the Sacramento and San Joaquin valleys there is an irrigated area of 250 000 acres and a population estimated at 300 000 where the population 15 years ago did not exceed 60 000.

42 As an example of what is being accomplished in Colorado, the following figures may be of passing interest to us and of value to the Irrigation Engineer.

43 The most striking feature of the conservation of rainfall is the impounding of the surplus water of one season against the deficiency of another. The work of the Western pioneers has accomplished wonders in making waste places habitable and in building up a territory otherwise barren and unproductive. The Reclamation Service has done good work in opening up for settlement a large amount of land. Table 16, compiled from the Governments records, shows the work in hand.

44 The work accomplished by the Reclamation Service is a good example of the conservation idea properly applied, and the example should be followed by States and individuals where practicable. The work of the Government in this line can not be over-estimated; it adds to our resources and that in a way that puts the land in the hands of the farmer.

Irrigation Statistics of Colorado 1906

TABLE 12 EXTENT OF IRRIGATION

	Division No. 1	Division No. 2	Total
Length of Main Ditch, Miles	3314	2840	6154
Length of Laterals, Miles	1927	2147	4074
Water used, Acre feet	1,228,821	1,607,292	2,836,113
Crops Irrigated			
Alfalfa	225,495	136,464	388,959
Natural Grasses	204,907	49,424	254,331
Cereals	328,357	67,051	395,408
Orchards	12,786	14,453	27,239
Market gardens	18,132	2,962	21,094
Potatoes	43,748	799	44,547
Sugar beets	81,375	46,668	128,043
Other crops	34,937	31,913	66,850
Total acres irrigated	921,675	376,734	1,298,409
Total acres that can be irrigated	1,310,125	533,737	1,843,862
Average water applied			
Acre feet per acre			

TABLE 13 YIELD OF IRRIGATED LAND

Boulder County

Fruit	Average yield per acre	Average selling price	Crop value per acre
Apples	300 boxes	$0.90 per box	$270
Cherries	300 crates	2.00 per crate	600
Blackberries	3500 quarts	0.15 per quart	525
Raspberries	4800 quarts	0.12 per quart	596
Strawberries	5000 quarts	0.10 per quart	500
Denver City and County			
Apples	300 boxes	$0.90 per box	$270
Plums	500 crates	0.50 per crate	250
Cherries	350 crates	2.00 per crate	700
Blackberries	1800 quarts	0.20 per quart	360
Raspberries	4000 quarts	0.15 per quart	600
Strawberries	2000 quarts	0.10 per quart	200
Jefferson County			
Apples	300 boxes	$0.50 per box	$150
Plums	400 crates	0.50 per crate	200
Cherries	350 crates	2.00 per crate	700
Blackberries	1600 quarts	0.20 per quart	320
Raspberries	3200 quarts	0.10 per quart	320
Strawberries	2400 quarts	0.10 per quart	240

These statistics are from the report of Thos. W. Jaycox, State Engineer.

TABLE 14 SUGAR BEET RECORDS

1907

Acres	Net Tons of beets	Tons per acre	Value of crop	Crop value per acre
16.0	264.70	16.6	$1329.80	$83.11
20.7	393.70	19.0	2000.79	96.65
17.8	541.50	30.4	2633.11	147.93
52.7	1250.00	23.9	6299.57	119.55
30.2	458.60	15.2	2303.44	76.30
9.7	258.40	26.6	1291.79	133.17
48.1	1025.50	21.3	4996.88	103.89
35.5	542.40	15.3	2711.77	76.39
24.0	485.00	20.2	2425.00	101.04
25.0	521.00	20.8	2655.28	106.21
47.0	877.60	18.7	4466.75	95.04
35.0	1028.70	29.4	5143.43	146.95
10.0	301.00	30.1	1504.61	150.46
338.0	5975.60	17.6	29877.90	88.40
47.2	828.40	17.6	4022.85	83.75
17.0	444.10	26.1	2218.99	130.53
37.0	816.70	22.1	4143.73	112.00
50.0	1012.30	20.2	5061.45	101.23
860.9	17025.20	391.10	$85087.14	$1952.60
	Average	21.7		108.46

Average cost of producing is about $35 per acre.

TABLE 15 CROPS WITHIN DRIVING DISTANCE OF DENVER

1907

	Acres	Yield	Selling Price	Value of crop per acre	Net per acre
Strawberries	14	1600 crates	$1.40 per crate	$588.00	
	25	1600 crates	2.00 per crate	640.00	$464
Raspberries*	12	500 crates	2.25 per crate	562.50	.....
	24	1000 crates	1.50 per crate	375.00	.....
	35	1600 crates	1.50 per crate	500.00	.....
Cabbage, average crop†	.....	20 tons	0.75 per hundred	300.00	.....
Onions	.....	.....	1.50 per sack	300 to 450	180 to 270
Late cauliflower	12	.....	.025 per pound	300.00	200
Celery	1	3000 dozen	.27 per dozen	810.00	600
Turnips	1	20 tons	12.00 per ton	240.00	165 to 190
Parsnips	1	20 tons	15.00 per ton	300.00	250

* Average cost of picking, crating and marketing $0.55 per crate

† Average cost of production, marketing, etc., $60 per crop

These statistics are from the report of Thos. W. Jaycox, State Engineer.

TABLE 16 AREAS, COST, EXPENDITURES, ETC., ON ENTIRE PROJECTS OR ON SUCH UNITS AS IT IS EXPECTED TO COMPLETE BY 1911

Location	Project	Area in acres	Estimated cost	Estimated expenditure to December 31, 1907	Per cent of completion
Arizona	Salt River	210,000	$6,300,000	$4,362,100	69.2
California	Orland	30,000	1,200,000	16,900	1.4
California, Arizona	Yuma	100,000	4,500,000	1,876,700	41.7
Colorado	Uncompahgre	140,000	5,600,000	2,900,000	51.8
Colorado	Grand Valley	50,000	2,250,000	9,750	0.4
Idaho	Minidoka	160,000	4,000,000	1,839,700	46.0
Idaho	Payette-Bois	100,000	3,000,000	1,381,500	46.5
Kansas	Garden City	8,000	350,000	282,000	80.5
Montana	Huntley	30,000	900,000	796,400	88.4
Montana	Milk River, including Saint Mary	30,000	1,200,000	314,800	26.2
Montana	Sun River	16,000	500,000	344,100	69.0
Nebraska, Wyoming	North Platte	110,000	3,850,000	2,797,300	73.0
Nevada	Truckee-Carson	170,000	4,800,000	3,804,600	79.2
New Mexico	Carlsbad	20,000	640,000	579,400	81.5
New Mexico	Hondo	10,000	370,000	358,600	97.0
New Mexico	Leasburg	10,000	200,000	167,900	83.9
New Mexico, Texas	Rio Grande	160,000	8,000,000	53,200	0.6
North Dakota	Pumping, Buford, Trenton, Williston	40,000	1,240,000	519,600	41.9
Montana, North Dakota	Lower Yellowstone	66,000	2,700,000	751,850	64.9
Oregon	Umatilla	18,000	1,100,000	765,500	69.6
Oregon, California	Klamath	120,000	3,600,000	1,305,080	36.2
South Dakota	Belle Fourche	100,000	3,500,000	1,281,900	36.6
Utah	Strawberry Valley	30,000	1,500,000	418,700	27.9
Washington	Okanogan	8,000	500,000	372,180	74.4
Washington	Sunnyside	40,000	1,600,000	481,180	30.7
Washington	Teiton	24,000	1,500,000	565,420	37.6
Washington	Wapato	20,000	600,000	5,220	8.7
Wyoming	Shoshone	100,000	4,500,000	2,313,990	51.5
		1,910,000	$70,000,000	$30,665,570	

From Blanchard

45 Irrigation projects are not proper subjects for the speculator. While there is a large increase in values resulting from a properly executed irrigation undertaking, the increase in value is not an "unearned increment:" the farmer must earn it. The fact that irrigation works are necessary for cultivation does not change the status of the land from that of farm land in general. The real development and improvement must be made by those fitted to till the soil and become permanent inhabitants of the reclaimed land. The inability

to see this simple truth has caused numerous speculators to lose money and has thrown a large number of highly advertised irrigation schemes into bankruptcy. Speculative interests will take desperate chances and trust to the ability of sales agents and glaring advertisements to "unload" before the inevitable failure comes. The reclamation act has avoided this rock, on which so many private undertakings split, by limiting the amount of land to each settler, and the cost per

TABLE 17 GRAZING, WOODLAND, FOREST, DESERT AND IRRIGATED LAND, AND EXTENT OF WATER SUPPLY IN THE WESTERN PUBLIC LAND STATES

States and Territories	Millions of Acres							
Arizona	72	38	14	5	15	0.2	0.2	2
California	99	20	19	19	20	15.0	1.5	17
Colorado	66	40	9	5		2.0	1.2	8
Idaho	54	20	12	11		1.0	0.5	5
Montana	93	56	7	12		1.0	0.8	11
Nebraska	49	25	2			22.0		2
Nevada	70	42	6	1	20	1.0	0.5	2
New Mexico	78	57	12	4		0.5	0.2	4
N. Dakota	45	38	1			6.0		2
Oregon	60	18	9	19		5.0	0.3	3
S. Dakota	49	38	1			10.0		2
Utah	52	18	16	4	10	2.0	0.5	4
Washington	43	9	13	13		2.0	0.1	3
Wyoming	62	39	3	4	5	1.0	0.5	9

TABLE 18 SWAMP LANDS

States	Area in square miles	States	Area in square miles
Florida	29,000	Indiana	1,250
Louisiana	15,000	New Jersey	900
Western States	10,000	New Hampshire	600
Arkansas	9,000	Massachusetts	500
Mississippi	9,000	Maryland	500
Michigan	7,500	Iowa	400
Minnesota	6,000	Vermont	400
Wisconsin	4,500	Nebraska	400
Maine	4,000	North Dakota	375
North Carolina	3,750	South Dakota	375
Georgia	3,750	Kentucky	350
Illinois	3,500	Pennsylvania	300
Missouri	3,000	Kansas	250
South Carolina	2,750	Connecticut	100
New York	2,500	Delaware	50
Alabama	1,750	Rhode Island	30
Virginia	1,600		
Tennessee	1,250		125,880
Ohio	1,250		

acre of the project, and by requiring the actual use and improvement of the land to acquire title. The act recognizes the fundamental principle of appropriation in the following language:—"That the right to the use of water acquired under the provisions of this act shall be appurtenant to the land irrigated, and beneficial use shall be the basis, the measure and the limit of the right." This provision of the act has been carefully drawn for the protection of the farmer and the confusion of the speculator.

46 The States that have lands subject to their control should follow this good example by adopting laws controlling the development of irrigation projects and exercise stringent control and supervision over water rights and districts subject to floods. In fact, the General Government should assume supervisory control over all projects for the development of lands liable to disastrous flooding. Table 17, taken from the Government records, shows in condensed form the general situation and the work remaining to be done.

SWAMP LANDS

47 Another field of operation is the reclamation of swamp lands. A little work is in progress in this line, but compared with the total amount of work to be done, it is not worth mentioning. Table 18, compiled from records of the Geological Survey by Herbert M. Wilson, shows the field for work.

POWER—WATER

48 In connection with irrigation and municipal water supply in arid and semi-arid regions is the closely related problem of power. In all the states in which appropriation of water is made, the appropriation for power purposes is recognized. As a general proposition, the development of a water-shed for irrigation and power purposes, jointly, is not practicable. In sections of the country where coal is cheap, the power phase of the water supply question will lie dormant until the cost of coal reaches a point that makes the installation of hydro-electric plants feasible from a money standpoint. There seems to be a hypnotic power attached to hydraulic power schemes that causes promoters and investors to lose sight of the fact that the real criterion is fixed charges *versus* operating expenses. It may also be the idea that they are going to get something for nothing which warps

judgment. Be that as it may, the fact remains that over-enthusiasm in the direction of hydraulic power plants has caused much trouble and loss of money. In locations such as the Pacific Coast, with coal at about $14 per ton, the hydro-electric plant finds its true habitat, but in the vicinity of coal mines, where cheap fuel prevails, the steam-driven plant is the present solution. So far as the conservation of water power is concerned, much may be said, but the problem, involving as it does the fundamental question of ownership by the individual, the state or the nation and the further legal question of riparian rights as opposed to appropriation, is one that it will require much study and many legal battles to solve.

49 The water power problem does not form a part of the conservation problem. The question is not one of the waste of water and the resource does not fall into either of the classes into which the President divided our natural resources. It is neither renewable nor non-renewable and its use does not diminish the supply. The question involved is one of ownership and the real conservation will be effected by legislation that will protect the public from the loss of valuable rights. The improvement in long distance transmission from the 10 miles of 10 years ago to the 200-mile lines of to-day makes water power available over large districts. In California four power companies, with an aggregate capital of some fifty millions of dollars control thirty hydro-electric and steam power plants. In the entire state about 250 000 h.p. is in operation under private control.

50 Mr. Lindsay states that there is a potential development of 800 000 h.p. on four rivers in the northern part of California that are under private control and that but 20 000 is utilized, the balance being tied up by speculative interests. Heretofore Congress has required no compensation for water power grants and during the last ten years has handed over to promoters and speculators 33 such grants *gratis*. The same authority states that of about 16 000 000 h.p. now in use in the United States, less than ¼ is from water power and 1 600 000 h.p. is going to waste over Government dams. The total water horse-power available for immediate use is given as 25 000 000.

51 I am well aware that the Waterways Commission was "tacked" on to the Conservation movement, but that was because Congress and the President had some differences of opinion regarding appropriations, and the water power question was also included. A strong veto message would stop a water power scheme for which the grantees were not required to render a just equivalent, without the consideration of a National Commission on Conservation of Natural Resources.

52 So far as our Western rivers are concerned, time has shown that the method of transportation which gives the lowest total cost of moving freight is the one that gets the business. For short hauls between river towns water transportation is cheaper and quicker than railways, for ordinary freight, but the advantage disappears as the distance increases. For export shipments, from St. Louis, the additional expense for insurance and interests on cargoes, with the cost of returning empty barges, has proved too great a handicap for river transportation to overcome. These items of expense are not materially lessened by river improvement, and it is not readily apparent how any depth of channel will materially lessen them or reduce fixed charges.

53 As the country becomes more thickly settled our rivers will have to be improved and controlled, and the sooner this work is inaugurated on proper lines the less will be the total cost of the work, but to put forth unsound arguments and to use the public works of the country to influence votes is not conducive to the continued development of our Republic. There is, at present, a strong tendency towards bureaucratic development that is inimical to the successful continuity of our form of Government. The waterways movement is not free from political influences and The American Society of Mechanical Engineers will do well to keep all political questions out of its proceedings.

POWER—COAL

54 Following logically after water-power is the problem of conserving our present great source of light, heat and power, *coal*. To the discovery and use of coal is to be attributed, more than to any other cause, the rapid development of the United States. It is the primary source of the power that has moved the wheels of commerce and manufacture. The rate at which we are consuming this heritage from Nature is sufficient evidence of the necessity of an inquiry into the probable supply.

55 Mr. Wilson, of the U. S. Geological Survey, estimates that our present rate of increase in the consumption of coal indicates that our supply will be exhausted in about 200 years. Estimates in this field, as in other long range predictions, are to be taken, however, as an integration of our present available differential coefficient. The law of the curve may not be properly expressed by deductions from our present knowledge, and future observations may prove that we are

drawing conclusions from insufficient data, obtained at the apex of the curve. I well remember that our Professor of Mining Engineering explained to the class of which I was a member the uselessness of prospecting for coal in the Rocky Mountains. It is quite plain at this time that his data were insufficient. We now realize that the conclusions of a Conference on the Conservation of Whales, had one taken place in New Bedford some years ago, would have created unnecessary alarm and expense. While whales were an important item in that day and generation, the present civilization does not need them.

56 The coal problem is, however, one requiring some consideration. Coal once consumed can not be replaced, and the steadily increasing cost of fuel must be faced. We have in the United States prices ranging from less than $1 to over $14 a ton for coal. I know of a railway power house, in Illinois, where the coal costs about $0.70 a ton delivered on the grates. The only way to conserve our supply of coal is to use it as economically as possible.

57 In the economical use of coal we have two main questions, mining and burning. With our easily mined Western coals it pays to use "wasteful methods" and these mining methods will continue until there is a radical change in present conditions. The mine operator and his engineer must meet competition. They do not adopt mining methods in advance of the times for fear of the certain end of their coal mining operations in bankruptcy. Just so long as it pays to work a mine for the best coal only, this process will continue and the future will have to take care of itself. Which one of you, as householder or engineer, will put up with a poor run of coal, in order that posterity may have good coal? The departments of our Government demand the best grade and are not willing to take the "run of the mine." This means that the poor grades are pushed on smaller purchasers or "go over the dump." If the Government wishes to set an example in the conservation of the coal supply, let an effort be made to adjust the power plants and heating plants in Washington to burn the low grade fuel.

58 Why does not the doctrine of "first in time is first in right," so strenuously introduced and so vigorously supported in the case of "water rights," apply, and permit the present generation to mine as they choose and grant the same privilege to succeeding generations? Who knows but that in the development of the plans of the "Supreme Architect of the Universe" worlds and planetary systems have birth, youth, manhood, old age and death and that we are but a small part

of the ways and means for executing a very small part of the universal plan? Our boasted civilization and rapid progress may be an indication that the end is rapidly approaching.

59 The outlook for improvement in mining, so far as conserving the coal supply is concerned, without a consolidation of competitors is not promising. How to accomplish the desired result under our form of government with unlimited competition, is not readily apparent; and mining will continue as in the past, until the Statesman and law maker, in deference to the opinions of the people, make radical changes in our fundamental laws. The present waste of both coal and life, in mining, should be reduced to a minimum. Both wastes are due, primarily, to the same cause. The selling price of coal, unless trust-controlled, must be kept down to competition prices, and in the Illinois mines the miner takes desperate risks in order to reduce his percentage of dead work, and occasionally pays for the risk with his life. The unnecessary loss of life has been reduced in other countries and can be reduced in our country. It is due to an uncontrolled greed for gain. Personally, I am of the opinion that the doctrine of "appropriation and beneficial use" offers many advantages not now understood and appreciated, and that it should be, so far as practicable, applied to our remaining natural resources under public control.

60 The Mechanical Engineer is more interested in the use than in the mining of coal, but here again he is confronted with the ever-present problem of "making both ends meet" and while he can not afford to be far behind the times, he realizes that he, also, can not afford to lose money by getting too far ahead of the times. Our last 20 years of progress in electrical lines is a good illustration of the cost of a rapid improvement in the general state of an art, and many of our electrical undertakings are loaded with fixed charges which represent the sins and shortcomings of preceding plants, relegated to the scrap pile before they had earned a fraction of their cost.

61 For the production of power the gas engine and producer give promise of a large saving in coal. The best data at hand indicating the saving that may result in ordinary practice are in the report of tests conducted at the World's Fair of 1903 by the U. S. Geological Department. The condensed results of the tests are shown in the following tables, which represent as nearly as possible the conditions of ordinary practice.

62 At present the gas engine and gas producer are in the stage of development. In the slang of the engineer, there are still some "bugs" to be gotten out of the apparatus. As applied to blast fur-

TABLE 19 COMPARATIVE SUMMARY OF LEADING RESULTS OF COAL TESTS MADE UNDER THE BOILER AND IN THE GAS PRODUCER

Name of sample	Duration of Trial		Total Dry Coal Consumed per Hour*		Dry Coal Burned per Square Foot of Grate Surface per Hour	
	Steam plant	Gas producer plant	Steam plant	Gas producer plant	Steam plant	Gas producer plant†
	Hours	Hours	Pounds	Pounds	Pounds	Pounds
Alabama No. 2	10.02	43.00	874	328.7	21.54	7.78
Colorado No. 1	9.97	30.00	722	341.7	17.80	7.56
Illinois No. 3	10.13	30.00	861	356.7	21.23	8.41
Illinois No. 4	10.02	30.00	938	348.5	23.13	7.96
Indiana No. 1	9.93	29.67	908	384.3	22.39	9.08
Indiana No. 2	10.13	7.00	832	312.0	20.51	7.13
Indian Territory No. 1	9.75	31.00	778	374.0	19.17	8.95
Kentucky No. 3	10.07	30.00	882	381.2	21.75	8.92
Missouri No. 2	9.98	4.33	1014	339.6	25.00	7.96
W. Virginia No. 1	9.98	24.00	768	315.6	18.94	7.36
W. Virginia No. 4	10.00	9.00	770	258.2	18.98	5.96
W. Virginia No. 9	10.00	6.33	721	320.1	17.78	7.60
W. Virginia No. 12	10.13	30.00	719	300.5	17.68	6.92
Wyoming No. 2	9.95	30.00	1075	416.5	26.51	9.50

Water Evaporated from and at 212 deg. fahr. per pound Dry Coal	B.t.u. per pound of Dry Coal Used		Electric Horse Power Delivered to Switchboard		Total Dry Coal per Electrical Horse Power per Hour*	
Steam plant Pounds	Steam plant	Gas producer	Steam	Gas producer plant	Steam plant Pounds	Gas producer plant Pounds
8.55	12,555	13,365	213.7	200.6	4.08	1.64
7.21	12,577	12,245	149.1	200.2	4.84	1.71
8.04	12,857	13,041	198.1	199.6	4.34	‡1.79
7.27	12,459	12,834	195.4	198.4	4.80	‡1.76
8.45	13,377	13,037	220.0	199.9	4.13	‡1.93
8.02	12,452	12,953	191.0	201.0	4.35	‡1.55
8.64	12,834	13,455	192.3	204.0	4.04	1.83
8.27	13,036	13,226	208.9	200.5	4.22	‡1.91
7.08	11,500	11,882	205.6	198.6	4.93	‡1.71
8.95	14,198	14,396	196.7	200.4	3.90	1.57
9.65	14,002	14,202	212.5	199.7	3.62	1.29
10.90	14,616	14,580	208.2	201.1	3.46	1.59
9.90	15,170	14,825	203.6	199.8	3.53	‡1.50
5.92	10,897	10,656	182.0	201.2	5.90	2.07

* In gas producer plants this includes the coal consumed in the producer and the coal equivalent of the steam used in operating the producer.

† Coal actually consumed in producer only.

‡ Gas producer hopper leaked during these tests.

nace work and for running on natural gas the gas engine is giving results that indicate its ultimate success. In furnace work the gas engine uses about 40 per cent of the blast furnace gas consumed under boilers supplying steam to a first-class steam-blowing engine of the same capacity. This rate is confirmed by data from a gas engine driven dynamo and indicates that a saving of about 60 per cent in coal is possible by substituting gas engines for steam engines of the ordinary type. On natural gas the guaranteed performance of the gas engine is generally at the rate of a horse power on 10 000 B.t.u. This with "12 cent" gas, in the Joplin lead district of Missouri, will make the fuel cost per kw-hr. on the bus 0.15 cents.

63 With a going concern the operating cost of producing power is generally a small part of the total cost, and the coal cost is still smaller, so that such factors as fixed charges and reliability are very important in deciding the kind of power plant best suited to furnish power for the undertaking. The true solution is found when the cost of power per unit of output is a minimum and there is no diminution of output due to power-house causes. In designing plants, with a high labor and a low material market, the Engineer often finds the power plant which will produce a horse-power-hour with the smallest coal consumption to be the most expensive plant under operating conditions. I have in mind from my personal experience two illustrations of this. In one case a power plant and arrangement of machinery were used that would make the graduate Mechanical Engineer just from school smile in derision, and in the other case the power plant, put in at the request of the owner, who expressed a desire to assist in the improvement of the general state of the art, would delight the heart of a professor of thermo-dynamics. The relative amount of coal per horse-power in the two plants were about 4 to 1 yet the plant which used 5 lb. of coal per horse power caused competitors to consolidate for the purpose of reducing expenses while the plant that used 1¼ lb. of coal per horse-power fulfilled the desire of the owner, who is still contributing to the advancement of the art and boasting of coal economy. It would not surprise me, however, to get an order at any time to put in a steam plant and dump the gas plant in the scrap pile.

64 In the West our labor market is not yet abreast of the best steam engine practice, to say nothing of the gas engine, and we can only wait until high cost of material shall enable the labor market to furnish the skilled labor necessary to operate the higher grades of power plants.

65 With true British perversity, James Watt adopted 33 000 ft. lb per minute as the unit of power, and use has fastened this ungainly number on the profession. For this unit we have a good running mate in the German fahrenheit degree. The British thermal unit is the amount of heat required to raise the temperature of 1 lb. of water 1 deg. fahr. This amount of heat is equivalent to about 778 ft. lb. of energy, so that for practical purposes we have 2545 B. t.u. as the equivalent of our unit of work, the horse-power-hour.

66 A pound of coal, ranging from lignites to the best steam coal, will develop an amount of heat ranging from about 9000 thermal units for the lignites to 15 000 thermal units for the best steam coals. If we could utilize all the energy stored in the coal we could produce from 3.54 to 5.89 h.p.hr. per pound of coal.

67 With the steam power plant as a means of transforming the potential heat energy of the coal into power we have, first, the losses due to the boiler plant, which with the most efficient plants will run at least 20 per cent and from that up to 50 per cent and more. Under every-day conditions a loss of 40 per cent may be expected. The next loss is due to the use of steam as a working medium. The theoretical efficiency possible is found by dividing the range of working temperature by the absolute temperature. With the best steam engines a thermo-dynamic efficiency of over 20 per cent is seldom obtained, so that, of the energy of the coal used in a steam power plant we can expect at the best no more than the equivalent of 15 per cent of the energy of the coal. The best efficiency to which I can personally certify is 16 per cent, and that with Illinois coal.

68 In one instance of good practice the heat delivered to the engine was accounted for as follows:

	Per cent
Converted into work	19.6
Radiated from engine	0.76
Rejected to condenser	74.6
Lost in drains from engine	1.71
Returned to boiler	3.33
Total	100.00

69 Good average conditions, however, may be stated for the entire plant about as follows:

	Per cent
Smoke stack gases	22
Radiation from boiler, etc.	7
Radiation from engine and pipe	4
Rejected to condenser	57
Converted into work	10
Total	100

70 Mr. H. G. Stott, Member of The American Society of Mechanical Engineers, estimates the average heat distribution in the power house as shown in Table 20. We may assume that with a good steam generating station we convert but ten per cent of the heat stored in the coal into electricity, on the bus-bars. Further losses due to distribution and conversion, in various ways, to light and power occur so that we get but little of the potential energy provided for our use by a wise and beneficent Nature. Surely, if our ecclesiastical brethren maintain that the storage of coal is a manifestation of Divine Providence, the present inventions for utilizing it must have emanated from his Satanic Majesty.

TABLE 20 AVERAGE HEAT DISTRIBUTION IN THE POWER HOUSE

	Per cent	Per cent
Heat in the coal		100
Loss in ashes	2.4	
Loss in stack	22.7	
Loss from boiler radiation and leakage	8.0	
Returned by feed water heater		3.1
Returned by economizer		6.8
Loss in pipe radiation	0.2	
Delivered to circulator	1.6	
Delivered to boiler feeder	1.4	
Leakage and high pressure drips	1.1	
Heating	0.2	
Loss in engine friction	0.8	
Delivered to small auxiliaries	0.4	
To house auxiliaries	0.2	
Radiation from engine	0.2	
Rejected to condenser	60.1	
Electrical losses	0.3	
Totals	99.6	109.9
Delivered to bus-bar		10.3

71 Another subject which is now engaging a great deal of the attention of advanced engineers is the process of converting coal into

gas in an apparatus called a producer and then burning the gas in the cylinder of a gas engine. This method gives a greater range of working temperature and consequently an outlook for greater efficiency of conversion of coal into work. With the first step of the process, however, the result will average in practice about the same as with the boiler, that is, there is a practicable efficiency of about 70 per cent. In the second step, that is, the conversion of the simple producer gas into power, the outlook for realizing the full benefit of the greater range of working temperature seems to be offset by the cooling necessary to maintain the working parts of the engine. The customary guarantee on gas engine performance is 1 h.p. on 11 000 B.t.u. or about 23.2 per cent efficiency, so that with simple producer gas we have about the same result as with the best steam engine, that is, about 15 per cent of the heat stored in the combustible converted into work.

72 It is quite safe to assume that with our best methods we can not expect more than 15 per cent of the heat value of the coal to be converted into work in the power house. With an efficiency of dis-

TABLE 21 LOSSES IN THE CONVERSION INTO ELECTRICAL ENERGY OF THE POTENTIAL HEAT ENERGY OF ONE POUND OF COAL

SELECTED EXAMPLES SHOWING THE AVERAGE FOR GOOD PRACTICE

With Steam Enriched Gas from Coal of 12 500 B.t.u.	Per cent
Loss in producer and auxiliaries	20.
Loss in cooling water in jackets	19.
Loss in exhaust gases	30.
Loss in engine friction	6.5
Loss in generator	0.5
Total losses	76.0
Converted into electrical energy	24.0

The Westinghouse Company Gas Engine Plant	Plant Per cent	Engine Per cent
Heat in coal	100.00	.....
Radiation	10.00	.....
Loss in ash	1.00	.....
Cooling gases	4.00	.....
Boiler fuel	10.00	.....
New heat to engine	75.00	100.00
Loss in jackets	26.25	35.00
Exhaust and radiation	27.20	36.30
Engine friction	2.82	3.75
Plant auxiliaries	1.50	2.00
Net work	17.25	23.00

TABLE 21—Continued

Results at the United States Geological Survey Testing Station	Per cent
Losses in producer and auxiliaries	29.0
Losses due to cooling gas, and apparatus	21.6
Losses in exhaust	30.2
Losses due to friction, etc	5.7
Delivered to bus bar	13.5

Analysis of the Buckeye Engine Co.	Per cent
Losses due to producer and auxiliaries	20.0
Losses in cooling water	22.0
Losses in exhaust	32.3
Losses in friction	3.6
Losses in dynamo	1.3
Delivered at switch board, electrical energy	20.8

tribution of 66 to 67 per cent, which, by the way, is good practice, we may get 10 per cent of the total energy of the coal to the customer, but if this is applied with an efficiency of 75 per cent, the customer realizes but 7.5 per cent of the stored energy of the coal. This is good engineering practice at present, and it is safe to say that but few plants are doing as well. We may safely assume the general average as not running over 5 per cent, or in other words we actually use but 5 per cent of the energy of the coal and waste 95 per cent, and, what is more, the outlook for improvement in this direction is not encouraging. In using electricity for lighting we get less than one per cent of the heat value of the coal as useful light. Of all our coal wastes, electric lighting is the greatest.

73 I am fully aware of the possibilities claimed for the gas engine, but my experience thus far, and the practical results with which I am conversant, do not indicate an early fulfillment of the claims. From present indications, the best steam and the best gas engine plants appear to be about on a par with regard to coal economy. The producer plant and the boiler plant are practically equal so far as the first step in converting the coal into power is concerned. The average gas engine is more efficient than the average steam engine, but the best gas engines and the best steam engines are about equally efficient.

74 The city of St. Louis replaced pumping engines that were in first-class running order, with high-duty pumping engines that reduced the coal bills about 75 per cent, but this was not done to conserve

the Illinois coal fields for the next generation. It was good engineering under the load conditions of a large pumping station but in many power plants the installation of the triple or quadruple expansion engine, while saving coal, would increase the cost of the output. I mention this case in order to impress on the rising generation of engineers that interest on first cost must be given due weight in designing plants in order to insure success.

TABLE 22 EXTENT OF THE COAL FIELDS OF THE UNITED STATES ESTIMATED BY THE UNITED STATES GEOLOGICAL SURVEY

States	Square miles	States	Square miles
Montana	47,200	Alabama	8,430
Texas	41,300	Indiana	7,290
Illinois	35,600	Utah	4,580
North Dakota	35,500	Tennessee	4,400
Missouri	23,000	South Dakota	2,400
Iowa	20,000	Virginia	2,120
Kansas	20,000	Arkansas	1,730
Wyoming	19,900	Washington	1,100
West Virginia	17,000	North Carolina	800
Kentucky	16,670	Maryland	510
Indian Territory	14,850	California	280
Pennsylvania	14,680	Oregon	230
New Mexico	13,500	Georgia	170
Ohio	12,660	Idaho	140
Colorado	11,600		
Michigan	11,300	Total square miles	388,940

TABLE 23 COAL OUTPUT FOR THE UNITED STATES FOR 1905

States	Short tons	States	Short tons
Texas	1,200,684	Tennessee	5,963,396
Utah	1,332,372	Kansas	6,423,979
Michigan	1,473,211	Iowa	6,798,609
Montana	1,643,832	Kentucky	8,432,523
New Mexico	1,649,933	Colorado	8,826,429
Arkansas	1,934,673	Alabama	11,866,069
Washington	2,864,926	Indiana	11,895,252
Indian Territory	2,924,427	Ohio	25,552,950
Missouri	3,983,378	West Virginia	37,781,580
Virginia	4,275,271	Illinois	38,434,363
Maryland	5,108,539	Pennsylvania	196,073,487
Wyoming	5,602,021		

TABLE 24 COAL OUTPUT OF VARIOUS COUNTRIES FOR 1905

Countries	Short tons	Millions of tons. Estimated supply. Summersbach, 1904
New Zealand	1,722,379	.....
S. A. Republics	2,968,117	.....
Spain	3,530,569	.....
New South Wales	6,742,186	.....
India	9,202,711	.....
Canada	8,775,933	.....
Japan	11,120,934	.....
Russia	21,294,639	40,000
Belgium	24,078,862	20,000
France	37,663,349	19,000
Austria-Hungary	45,209,933	17,000
Germany	191,576,074	415,000
Great Britain	264,464,408	193,000
United States	392,919,341	681,000

TABLE 25 CONSUMPTION OF COAL IN THE UNITED STATES

Year of census	Population	Decade	Coal Consumed	Pounds per capita per year
1820	9,638,453	1816 to 1825	331,356	6.9
1830	12,866,020	1826 to 1835	4,168,149	64.8
1840	17,069,453	1836 to 1845	23,177,637	271.6
1850	23,069,453	1846 to 1855	83,417,825	723.2
1860	31,443,321	1856 to 1865	173,795,014	1105.5
1870	38,558,371	1866 to 1875	419,425,104	2175.6
1880	50,155,783	1876 to 1885	847,760,319	3380.6
1890	62,622,250	1886 to 1895	1,586,098,641	5065.8
1900	75,568,686	1896 to 1905	2,832,599,452	7496.8

75 It will be noticed that the curve of the per capita rate takes on a very rapid rate of increase at 1860. The rate from 1870 to 1900 is approximately represented by the following formula:

$$\text{Log } C = 3.466 \ (\log T - 0.882)$$

in which T represents the time elapsed in years from 1800, and C represents the pounds of coal consumed per capita per year. The several values resulting from this formula are shown in Table 26.

76 Combining this formula with that previously given for the rate of increase of population, we obtain the following:

$$\left.\begin{matrix}\text{Log of annual consumption} \\ \text{coal in millions of tons}\end{matrix}\right\} = 5.168 \log T - 7.673$$

77 Now *if* the rate at present indicated continues, we may estimate our annual consumption of coal roughly, as in Table 27. This total consumption of 614 900 millions of tons of coal would about exhaust the total supply of the country, at the present rate, in the next 100 years. It is not probable, however, that this rate will continue, but that the curve will reach a maximum and then slowly recede, becom-

TABLE 26 CONSUMPTION OF COAL PER CAPITA IN THE UNITED STATES WITH ESTIMATE TO 1950

Year	Per capita consumption of coal per year Pounds	Per capita rate from census reports
1870	2176	2176
1880	3457	3381
1890	5194	5066
1900	7499	7497
1910	10,400	....
1920	14,090	....
1930	18,630	....
1940	24,040	....
1950	30,550	....

TABLE 27 ESTIMATED ANNUAL CONSUMPTION OF COAL IN THE UNITED STATES

Year	Rate millions of tons per year	Total tons consumed during the decade
1900	460	4,600
1910	750	7,500
1920	1,190	11,900
1930	1,790	17,900
1940	2,620	26,200
1950	3,430	34,300
1960	5,220	52,200
1970	7,460	74,600
1980	9,240	92,400
1990	12,750	127,500
2000	16,580	165,800
	Total	614,900

ing asymptotic to the time axis, when the coal is all consumed or a substitute is found, or something else happens. Prognostications based on the assumption that the population and the coal consumption curves will continue along the paths of the past 40 years must be introduced with a big If, and the if must be given due weight. The study of mathematics and the science of numbers develops a keen

sense of humor, unappreciated by the laity. The law of past increase may hold good for a short time, but it is quite certain that it cannot continue indefinitely. New conditions will obtain and new methods and materials will be introduced; alcohol manufactured from waste vegetable matter is one possible alternative. When our successors can no longer afford electric light and traction they will have to get along without it.

MINERAL RESOURCES

78 Of our mineral resources, iron stands at the head as the distinct feature in our past development along mechanical lines, notwithstanding that electricity is trying to relegate us to the "copper age." An unlimited supply of metal is not needed. We require simply sufficient metal for a working capital and to keep the stock good by supplying for wear and tear. At the present time an immense amount of iron is lying in the wrecks and scrap iron piles all over the country because it does not pay to work it over.

79 The latest estimates, based on visible supply and rate of consumption, indicate an exhaustion of the supply in about 200 years. We waste iron and steel because it is cheap, but as time goes on the waste will diminish and the use be curtailed as the law of supply and demand shall dictate. The most reliable estimate that I have been able to find, places the world's production of pig iron for 1907 at about 61 000 000 tons, of which the United States furnishes about 26 000 000. In 1872 the output of the United States was about 2 000 000 tons. These amounts give us a per capita rate of about 0.05 tons in 1872 and 0.30 tons in 1907. As to the gradually increasing price of iron we quote Mr. Frank A. Munsey to the effect that the Steel Corporation contracted with the Great Northern Railroad (James J. Hill) to take ore from the Great Northern, "the price to be advanced each year over the preceding year, 3.4 cents. The price for 1897 was $0.85 per ton."

80 The conservation of iron must be by reducing waste to a minimum. As the supply of iron decreases the price will increase and the scrap iron merchant will become an important factor. The mathematical limit appears to point to the control of the old iron market as the ultimate function of the Steel Trust of the future.

81 Of the conservation of oil and natural gas we have neither time nor inclination to discuss the profligate waste in this country, particularly in the South-West where wells have been bored only to

waste the raw petroleum products and the money expended in drilling. All are familiar with the natural gas history of Pittsburg, but the waste in that case is not to be compared with that of the great South-West. The operation of natural laws in the case of these resources has rendered conservation unnecessary in many localities. The only thing that can stop this brand of insanity is a strict application to this class of natural resources of the doctrine of appropriation and beneficial use. The entire community has an interest in natural resources and should not permit waste. This doctrine is not the idle dream of the socialist but is the result of years of experience and is probably the oldest fundamental law on the statute books of the world.

Conclusion

82 Individually the members of the Society will find it necessary in the practice of the profession to design, build and operate works so as to compete with others in the same line. The crucial test will always be found in the cost per unit of output, and the real work of the engineer, in the field of conservation, will be measured by the ability of the works designed by him to compete in the markets of the world. As times and conditions change the engineer must modify his plans and processes or meet with failure. The United States at present is a country of cheap materials and high-priced labor and the result is that it pays to waste material to save labor. In countries where the reverse is found, that is, cheap labor and high-priced materials, it pays to waste labor to save materials. This fact is brought home only to the engineer who has had the opportunity to design work under foreign conditions.

83 With our present high cost of labor we must take all possible precautions to keep the price of materials low. With high-priced materials and high-priced labor we will be shut out of the world's markets. In fact, as it now stands, other countries are under-selling our mills in our own country, by virtue of low-priced skilled labor. The true exponent of the conservation movement should be the skilled labor of the United States.

84 The engineer is also interested, as one of the producing class, in the conservation of the materials of the country and likewise in the conservation of the results of his own labor. The engineering societies must fall in with the conservation idea and see to it that the returns from the societies are commensurate with the effort expended in operating them. Our societies must change with the times and

adopt new methods as the occasion requires. The American Society of Mechanical Engineers has been alive to this fact, as the journals of the year will show. The credit of the work belongs to the Secretary of the Society, Mr. Calvin W. Rice, and his efficient corps of assistants.

85 In closing I take the liberty of suggesting that the Society take one more step along the road leading to the conservation of the personal funds of its members and at the same time extending its own influence and adding to its own exchequer. This step is the production of an Engineers Notebook that will give the practicing Mechanical Engineer the fundamental constants needed for daily use and enable him to throw several yards of so-called note books from his library. Such a book, published by the Society for the use of its members, would be an incentive to engineers to join the Society and save the young members of the profession enough money to pay for life memberships.

86 I venture to suggest that most of the older members have paid out more money for note books than for dues to the Society and I most earnestly recommend that The American Society of Mechanical Engineers make an effort to conserve both the time and the funds of future members by issuing a note book which shall gradually become the standard mechanical note book, which will be a fund of information for new members, and which will make the Society known wherever engineering is practiced.

THE TRANSMISSION OF POWER BY LEATHER BELTING

CONCLUSIONS BASED PRINCIPALLY ON THE EXPERIMENTS OF LEWIS AND BANCROFT

By Carl G. Barth, Philadelphia, Pa.
Member of the Society

In his paper, Slide Rules in the Machine Shop as a Part of the Taylor System of Management, read December 1903, the writer referred to an improved theory and new formulæ developed by him for the pulling power of belting, which had been applied in connection with the slide rules described. He also stated his expectation of presenting his theory and conclusions to the Society at some future time.

2 These conclusions have since been successfully applied in practice by the extensive daily use of these slide rules in task-setting for machine operations, and the present paper was prepared with the general view of submitting this theory to the Society for the criticism and consideration of members who are interested in this subject and with the special view of supplementing Mr. Taylor's paper, On the Art of Cutting Metals. All the experimental and mathematical data for the slide rules were presented in his paper, excepting the data upon the pulling power of belting, an important element in these slide rules when applied to belt-driven machines.

3 The theory to be presented is only an additional attempt, and it is hoped a fairly successful one, to do something along lines repeatedly touched by various other investigators among the members of the Society, and the writer is glad to acknowledge his indebtedness to nearly all of these, as his work has principally consisted in taking advantage either of carefully conducted experiments recorded by them, or of suggestions of various kinds that have stimulated his interest and set him thinking.

To be presented at the Monthly Meeting (January 12, 1901) of the American Society of Mechanical Engineers. All papers are subject to revision.

4 Most notable is the paper presented by Mr. Wilfred Lewis at the Chicago meeting in 1886, in which he recorded a series of experiments conducted by himself in the spring of 1885, under the direction of Mr. J. Sellers Bancroft and at the works of William Sellers and Company, Philadelphia. In his paper was shown for the first time the fallacy of the then universal and still common assumption, that the sum of the two tensions in a belt is constant for all loads. It was also shown that the coefficient of friction between a belt and its pulley is considerably higher than was commonly assumed for ordinary working conditions, and that this coefficient varies greatly with the velocity of slip, a fact that has also been pointed out by other investigators.

5 Mr. Lewis did not, however, even attempt to develop either empirical or rational mathematical formulæ to represent the facts that he established, though his experiments, as will subsequently appear, contained all that was necessary for a complete mathematical exposition of the subject.

6 An attempt at an empirical mathematical exposition of the relations between the two tensions in a belt in accordance with the facts established in Mr. Lewis' paper, was later made by Prof. Wm. S. Aldrich in a paper read at the New York meeting in 1898. Mr. Aldrich made an original layout of a great number of Mr. Lewis' experiments in a manner that seemed to the writer to indicate an excellent way to investigate the subject, and which resulted in a discussion of Mr. Aldrich's paper, the substance of which has formed the basis for all of the writer's subsequent work on the subject.

7 But while the writer's complete theory could have been worked out without it, its practical application to the running of belt-driven machine tools could never have been made in the present satisfactory manner without the facts made known by Mr. Taylor in his paper, Notes on Belting, read at the New York meeting in 1893.

8 In this paper Mr. Taylor showed the economy of running belts under much lower tensions than those commonly used, and that the ultimate strength of a belt, or rather of the joint in a belt, does not form a proper basis for the working tension of a belt, since a belt will not long retain a tension that is even a small fraction only of its ultimate strength (see Fig. 4). However, Mr. Taylor's facts and figures were derived from comparatively slow-running belts, and he gave nothing that could be directly applied to the higher and more economical belt speeds. The modification and extension of Mr. Taylor's ideas to include high speed belts have therefore been part of the writer's personal work also.

9 To sum up, the writer's work on the subject has mainly consisted of the following:

a To establish a mathematical formula for the relation between the tension in a belt and the stretch due to this tension, based on experiments made at different times by Mr. Wilfred Lewis, Prof. W. W. Bird and himself. See Fig. 1, 2 and 3, and Par. 1–18 of the Appendix.

b By means of the knowledge of the elastic properties of leather belting expressed by this formula to develop a formula for the relations between the tensions in the two strands of a belt transmitting power, which formula takes account of the influence of the sag in a horizontal belt, and agrees substantially with the results of the experiments made by Mr. Lewis, when plotted in the manner first done by Professor Aldrich. See Fig. 6 and Par. 19–38 of the Appendix.

c To establish a formula to express the relation between the coefficient of friction between a belt and a cast iron pulley, and the velocity with which the belt slips or slides over the pulley, as revealed by plotting the results likewise obtained by Mr. Lewis. See Fig. 7, and Par. 48 of the Appendix.

d The construction of a diagram embodying the formula expressing the relation between the two tensions in a belt, the well known formula for the loss in effective tension due to centrifugal force, and the likewise well known formula for the ratio between the effective parts of the two tensions, as determined by the coefficient of friction and the arc of contact of the belt on its pulleys. These formulæ are so correlated on the diagram that problems dealing with the contained variables may be solved graphically, while a direct algebraic solution is possible only for a vertical belt, or what is the same thing, by neglecting the effect of the sag of a horizontal belt. See Plate 1 and Par. 11–24.

e Also, by means of the better knowledge gained of the elastic properties of leather belting, to develop a formula for the creep of a belt on its pulley due to the difference in the tensions in the two strands, along the lines outlined by Professor Bird in his paper on Belt Creep, read at the Scranton meeting in 1905. See Par. 41–44 of the Appendix.

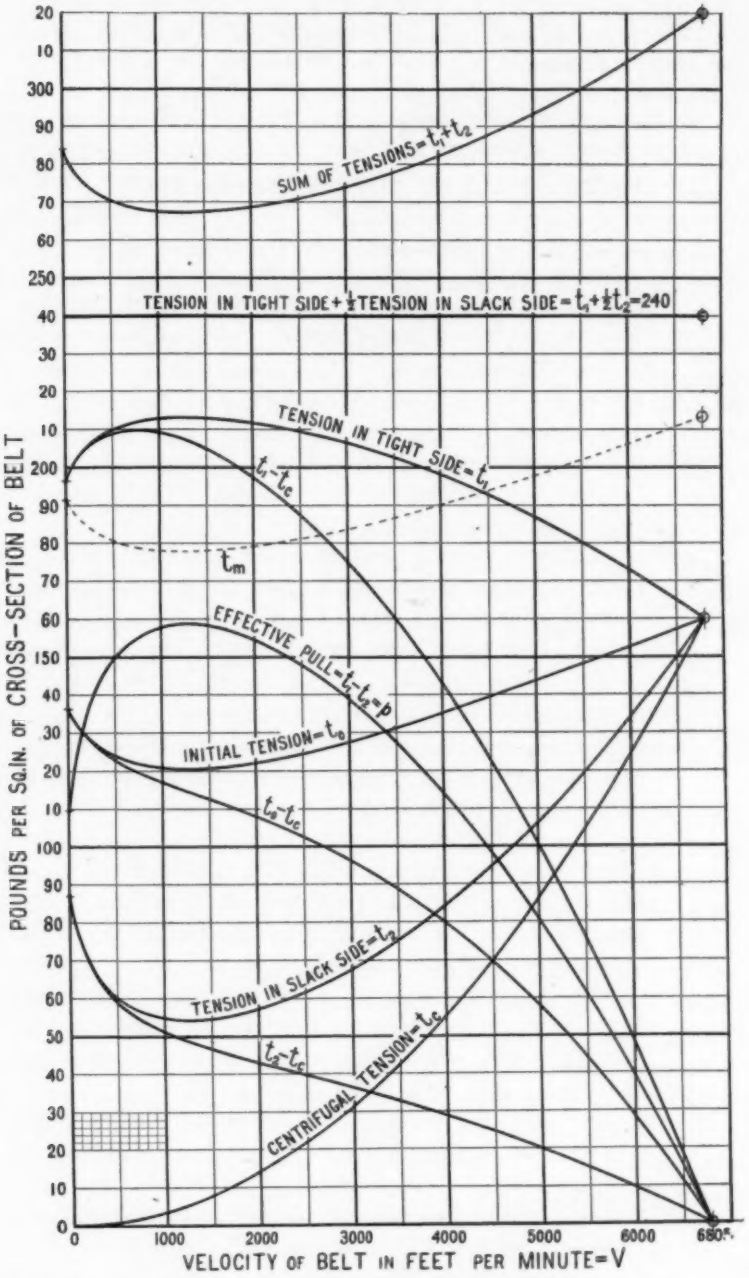


FIG. 1 DIAGRAM SHOWING THE RELATIONS OF PULLING POWER TO TENSIONS, AT ALL SPEEDS FOR A BELT OF 1 SQ. IN. SECTION

See opposite page

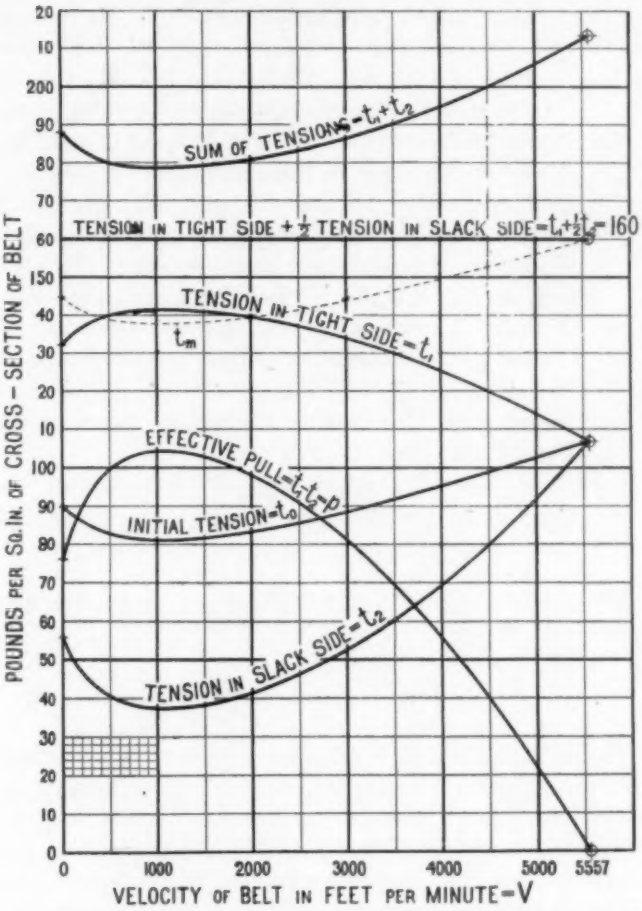

SUM OF TENSIONS = t_1+t_2
TENSION IN TIGHT SIDE + ½ TENSION IN SLACK SIDE = $t_1+\frac{1}{2}t_2$ = 160
TENSION IN TIGHT SIDE = t_1
t_m
EFFECTIVE PULL = $t_1-t_2=p$
INITIAL TENSION = t_0
TENSION IN SLACK SIDE = t_2
POUNDS PER Sq. IN. OF CROSS - SECTION OF BELT
VELOCITY OF BELT IN FEET PER MINUTE = V
0
1000
2000
3000
4000
5000
5557

FIG. 2 DIAGRAM SHOWING THE RELATIONS OF PULLING POWER TO TENSIONS AT ALL SPEEDS FOR A BELT OF 1 SQ. IN. SECTION

Plotted with same data as Fig. 1 except that here $A = 160$ lb. for the full-drawn curves.

In Fig. 1 arc of contact = 180 deg.; coef. of friction $f = 0.5A - \frac{140}{500 + V}$; and the sum of tension in the tight side of belt and one-half the tension in slack side is $A = 240$ lb. for all velocities of the belt. The dotted curve marked t_m gives initial tension that with p the same as that figured from $A = 240$ corresponds to $A = 320$. This is the *maximum initial tension* to which a *machine belt* is retightened whenever the same has fallen to the *miminum initial tension* t_0.

In Fig. 2 the dotted curve t_m gives initial tension that with p the same as that figured from $A = 160$ lb. corresponds to $A = 240$ lb. This is the *maximum initial tension* to which a *countershaft belt* is retightened whenever the same has fallen to the *minimum initial tension* t_0.

f The construction of diagrams showing the pulling-power and other relations of the two tensions of a belt of 1 sq. in. cross section and 180 deg. arc of contact at different speeds, under certain conditions and assumptions recommended by the writer. See Fig. 1, 2 and 3, and Par. 38–52. Also a modification of these diagrams for extended practical use, on which may be read off: (1) The pulling power of a belt of any width and thickness and any arc of contact, between 140 and 180 deg.; (2) The initial tensions below which the belt must not be allowed to fall in order to confine the slip and the consequent loss of efficiency of transmission within certain limits; (3) The initial tension to which it is recommended that the belt be re-tightened after falling to this minimum limit. See Plate 2 and Par. 53–66.

g Finally, the construction of a slide rule serving the same purpose as the diagram just mentioned, but which is much handier than the diagram. See Fig. 5.

10 With these statements the explanation of the diagram, Plate 1, will now be taken up.

DESCRIPTION AND USE OF THE DIAGRAM PLATE 1

11 Taking the extreme left-hand bottom corner point as the origin, distances along the bottom line represent the variable tension in the tight strand or side of a belt in terms of the initial tension, while vertical distances measured to any of the bottom group of curves in the middle field of the diagram represent the corresponding tension in the slack side of the belt, also in terms of the initial tension.

12 The particular curve against which to read off a certain tension depends on the center distance of the pulleys of the belt in connection with its initial tension per square inch, and is found by consulting the small diagram directly to the right of this group of curves, in the following manner:

13 Read off the center distance c along the extreme right side of this diagram, then follow along the diagonal to the left from this reading of c until it intersects the vertical line that extends up from the reading of the initial tension t_0 on the base of the diagram.

14 From this point of intersection of c and t_0 go horizontally to the left to the reading of the corresponding value of the ratio

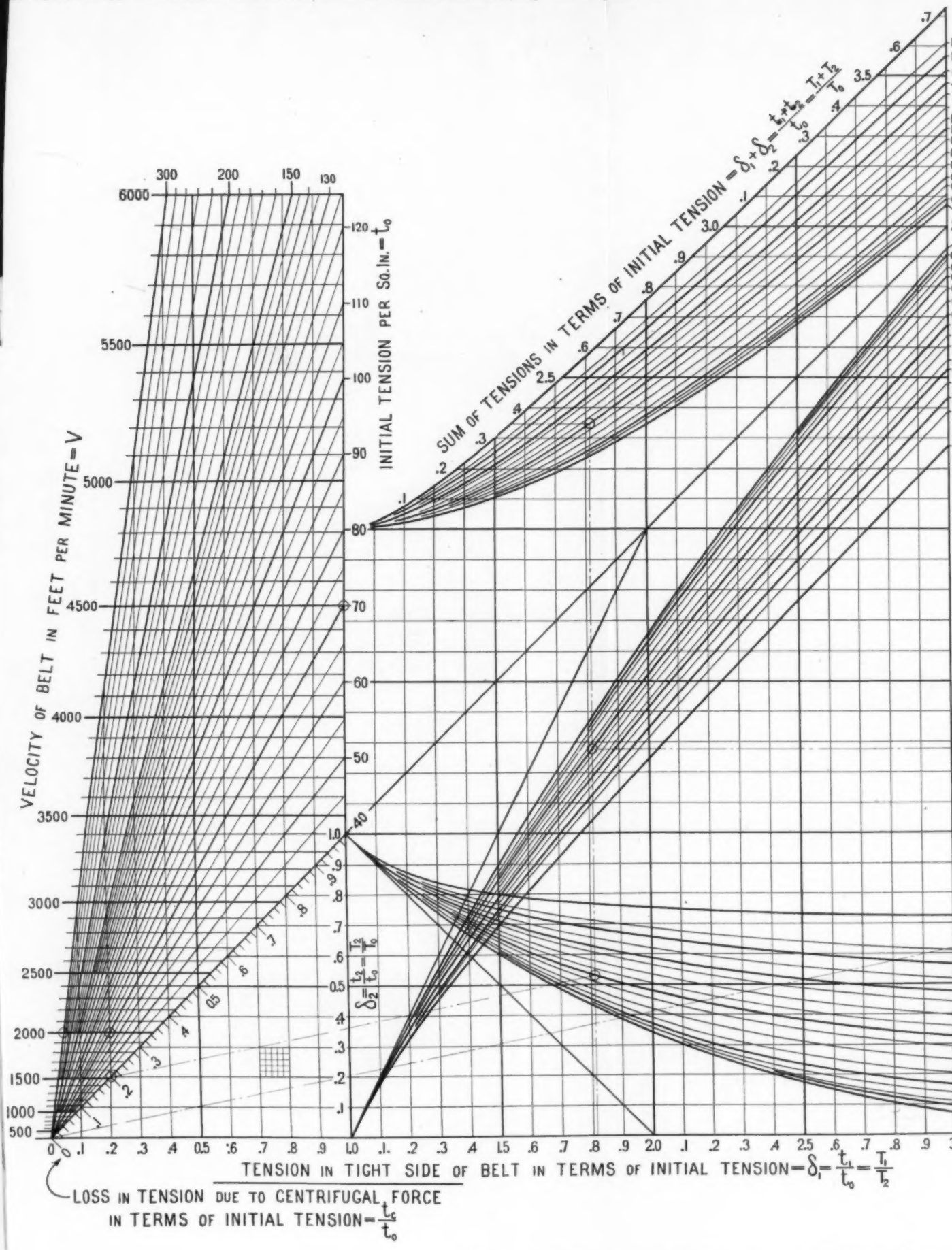
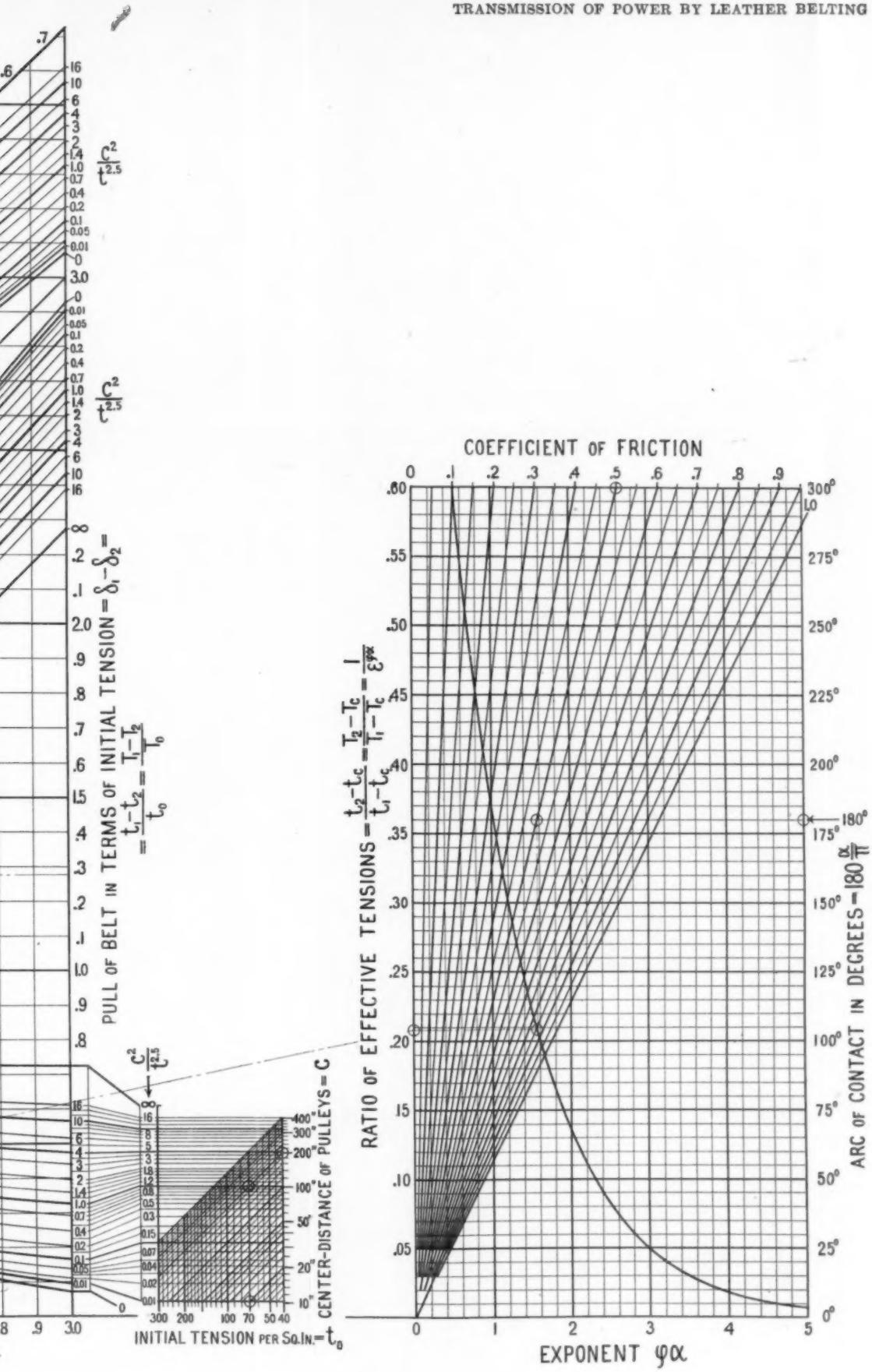

VELOCITY OF BELT IN FEET PER MINUTE = V
INITIAL TENSION PER SQ. IN. = t_0
SUM OF TENSIONS IN TERMS OF INITIAL TENSION = $\delta_1+\delta_2=\frac{t_1+t_2}{t_0}=\frac{T_1+T_2}{T_0}$
$\delta_2=\frac{t_2}{t_0}=\frac{T_2}{T_0}$
TENSION IN TIGHT SIDE OF BELT IN TERMS OF INITIAL TENSION = $\delta_1=\frac{t_1}{t_0}=\frac{T_1}{T_2}$
LOSS IN TENSION DUE TO CENTRIFUGAL FORCE IN TERMS OF INITIAL TENSION = $\frac{t_c}{t_0}$

PLATE 1 DIAGRAM FOR THE GRAPHICAL SOLUTION OF FORMUL


COEFFICIENT OF FRICTION
RATIO OF EFFECTIVE TENSIONS = $\frac{t_2-t_c}{t_1-t_c} = \frac{T_2-T_c}{T_1-T_c} = \frac{1}{\varepsilon^{\varphi\alpha}}$
EXPONENT $\varphi\alpha$
ARC OF CONTACT IN DEGREES = $180\frac{\alpha}{\pi}$
PULL OF BELT IN TERMS OF INITIAL TENSION = $\delta_1-\delta_2 = \frac{t_1-t_2}{t_0} = \frac{T_1-T_2}{T_0}$
$\frac{c^2}{t+2.5}$
CENTER-DISTANCE OF PULLEYS = C
INITIAL TENSION PER SQ.IN. = t_0

FOLDER No. 1

$\frac{c^2}{t_0^{2.5}}$ which leads directly to the proper curve in the bottom group of curves in the middle section of the diagram.

15 Against this curve there can now be read off any simultaneous tensions in the two strands of the belt corresponding to these particular values of c and t_0 of the belt under consideration.

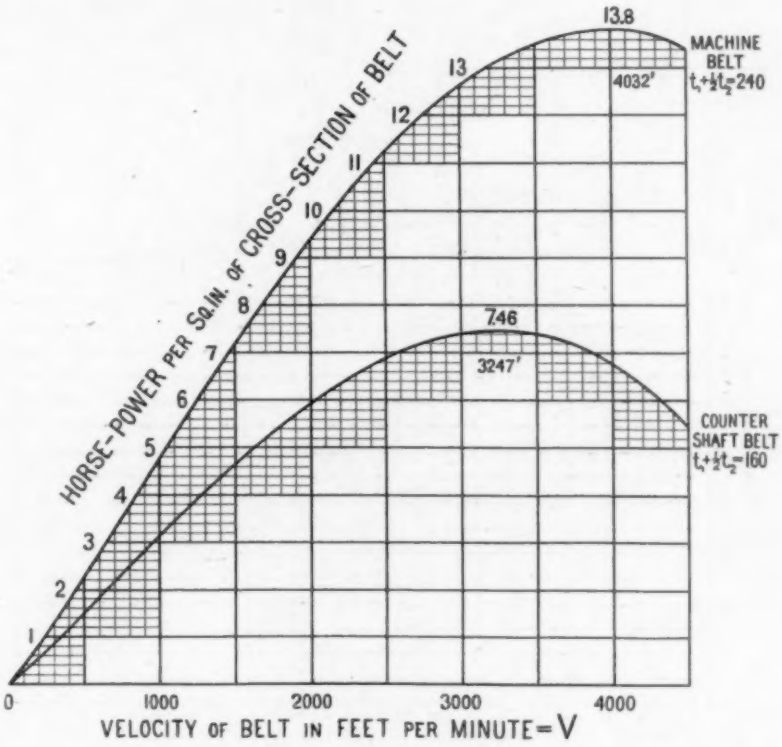
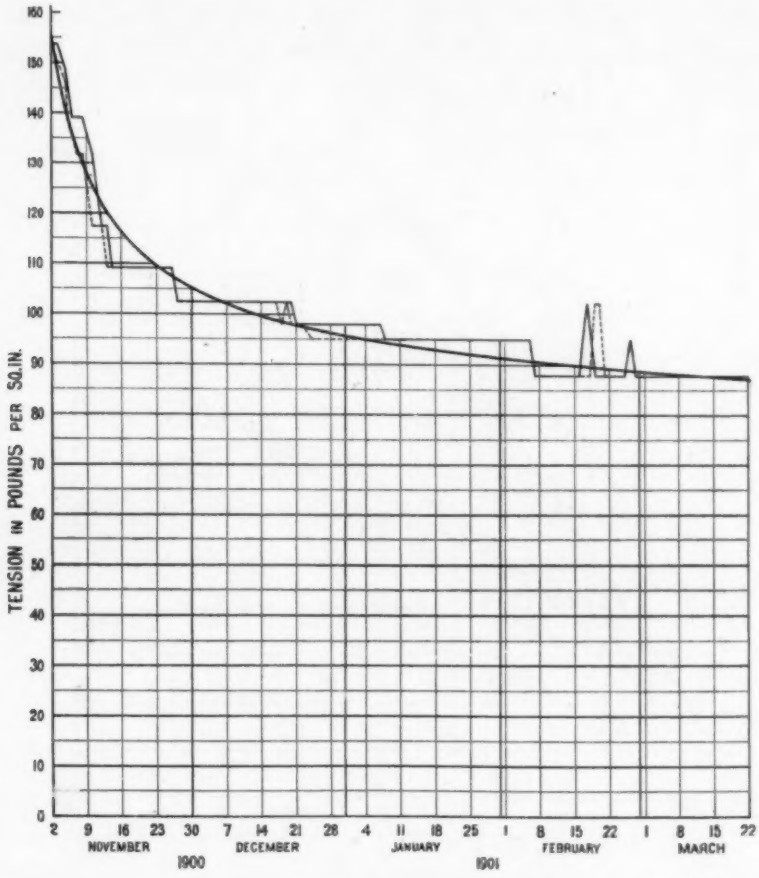


FIG. 3 HORSEPOWER OUTPUT CORRESPONDING TO BELT PULLS IN FIG. 1 AND 2

16 Having noted this curve, and turning to the extreme right hand section of the diagram, the ratio of the effective tensions in the two sides of the belt corresponding to the particular coefficient of friction ϕ, and the particular arc of contact α which we wish to count on, may be determined.

17 To this end we read off the arc of contact in degrees on the extreme right-hand side of this section of the diagram, follow this

reading horizontally to the left until it intersects that line radiating from the bottom left corner of this section which is marked with the value assumed for ϕ at its termination in the extreme top line of the



EXPERIMENTS MADE ON THE FALL IN TENSION IN TWO BELTS $5\frac{3}{4}$″ WIDE BY $\frac{13}{32}$″ THICK DRIVING A LARGE ROTARY PLANER AT THE WORKS OF THE BETHLEHEM STEEL COMPANY.

The peculiarly high tensions measured on four days, during the latter part of February 1901, were probably due to something sticking in the belt scales used.

section, and then from this point of intersection go vertically up or down as the case may be, until we meet the single curve drawn in this section of the diagram. From this point in the curve we now go

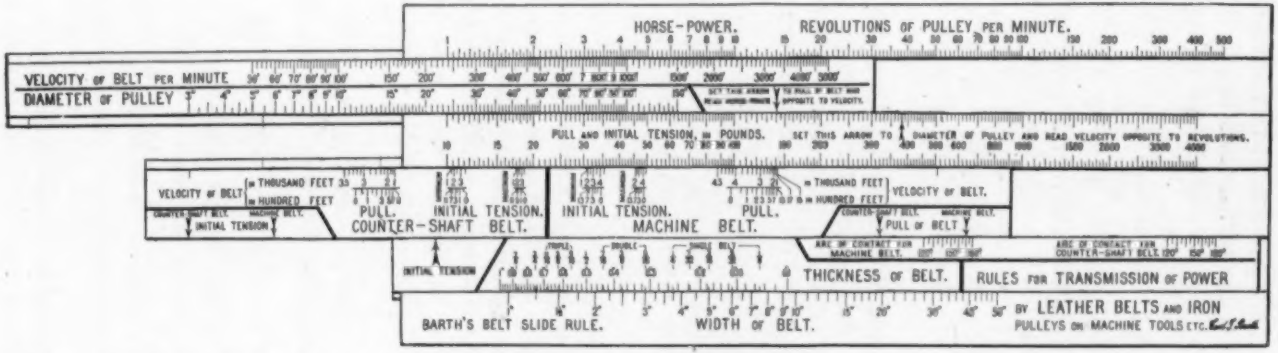


FIG. 5 SLIDE RULE FOR THE MORE CONVENIENT SOLUTION OF BELT PROBLEMS SOLVED BY THE DIAGRAM, PLATE 2, BASED ON FIG. 1 AND 2

horizontally to the extreme left side of this section of the diagram and there read off the value of the ratio of the effective tensions, which is

$$\frac{t_2}{t_1} = \frac{1}{e^{\phi\alpha}}$$

for a belt running so slowly that the centrifugal force has no perceptible influence, and equal to

$$\frac{t_2 - t_c}{t_1 - t_c} = \frac{1}{e^{\phi\alpha}}$$

when the centrifugal force reduces the total tensions to the effective tensions $t_2 - t_c$ and $t_1 - t_c$ respectively.

18 From the point representing $\frac{1}{e^{\phi\alpha}}$ we now draw a line to the extreme left bottom corner point of the whole diagram.

19 Any two simultaneous coördinates to this slant line counted from its extremity in the base line of the diagram, will then also be in the ratio $\frac{1}{e^{\phi\alpha}}$

20 Passing now to the extreme slanting left side of the diagram, we read off the velocity V of the belt, follow this reading horizontally to the right until we intersect that radiating line from the extreme left bottom corner point of the diagram which is marked with the initial tension t_0 of the belt per square inch where it terminates, either in the extreme top line of this section of the diagram, or against its extreme right side.

21 From this point of intersection we now go down vertically until we reach the long 45-deg. diagonal of the diagram, on which we then read off the ratio $\frac{t_c}{t_0}$ or the loss in effective tension in terms of the initial tension, due to the centrifugal force in the belt.

22 From this point on the long 45-deg. diagonal of the diagram we now finally draw a line parallel to the line previously drawn to represent the ratio $\frac{1}{e^{\phi\alpha}}$ and extend it to intersect the curve first of all determined to represent all possible relations between the two tensions in the belt.

23 The coördinates of this point then gives the two tensions in the belt in terms of its initial tension. By extending the ordinate up to intersect that curve in the middle group of curves which is marked with the same value of $\frac{c^2}{t_0^{2.5}}$ at its terminal in the right side of this middle section of the diagram as the curve dealt with in the bottom group of curves, we read off the difference of the two tensions; that is, the effective pull of the belt, in terms of the initial tension.

24 By likewise extending the ordinate all the way up to meet that curve in the top group of curves which is marked the same as the other two curves, we may also read off the sum of the two tensions, in terms of the initial tension.

25 *Example.* A belt of the pulley center-distance $c = 200$ in. and of $2\frac{1}{2}$ sq. in. cross section, has an initial tension $T_0 = 175$ lb. and runs at a velocity $V = 2000$ ft. per minute. The arc of contact of the belt on each pulley may be taken as 180 deg., and the coefficient of friction ϕ as 0.5. What will be the centrifugal tension $T_c = 2.5\ t_c$ in the belt, what the tension $T_1 = 2.5\ t_1$ in the tight side, and what the tension $T_2 = 2.5\ t_2$ in the slack side? Also what will be the effective pull $P = T_1 - T_2$ and what the sum $T_1 + T_2$ of the two tensions?

26 *Solution.* The steps have been indicated on the diagram by little circles around the points on the several scales of variables that correspond to the values of these variables in the example. Thus, in the small diagram at the bottom of the chart, between the other two diagrams, circles have been drawn around the points indicating the unit initial tension

$$t_0 = \frac{175}{2.5} = 70,$$

and the center distance 200, giving the resulting value

$$\frac{c^2}{t_0^{2.5}} = \frac{200^2}{70^{2.5}} = 1, \text{ approximately,}$$

which determines the particular curve in each of the three groups of curves in the middle section of the diagram which apply to the belt under consideration.

27 Also, in the right-hand section of the diagram circles have been drawn indicating the coefficient of friction $\phi = 0.5$, and the arc of con-

tact 180 deg., giving the resulting value of the ratio between the effective tensions $\frac{1}{e^{\phi\alpha}} = 0.208$.

28 From the point $\frac{1}{e^{\phi\alpha}} = 0.208$ has also been drawn a line to the extreme left bottom corner of the whole diagram, the ratio of any two coördinates to this line thus being 0.208 also.

29 Again, in the left-hand section of the diagram circles are drawn about points indicating a velocity $V = 2000$ ft. and the initial tension $t_0 = 70$ leading to a resultant value of

$$\frac{t_c}{t_0} = \frac{T_c}{T_0} = 0.202$$

This means that $0.202 \times 175 = 35.35$ of the total 175 lb. of initial tension in the belt, is made ineffective by the centrifugal force due to $V = 2000$ ft.

30 From the point $\frac{t_c}{t_0} = 0.202$ has also been drawn a line parallel to the line expressing the ratio

$$\frac{1}{e^{\phi\alpha}} = 0.208$$

so that the inclination of this line expresses the same ratio between the effective belt tensions; or

$$\frac{T_2 - T_c}{T_1 - T_c} = \frac{t_2 - t_c}{t_1 - t_c} = 0.208$$

31 The point of intersection of this line with the curve previously found to express all possible relations between the working tensions in the two strands, has also been encircled, and a vertical line has been drawn through this point upwards until it intersects that curve in the top group of curves which is marked $\frac{c^2}{t_0^{2.5}} = 1$, and which accordingly expresses all possible values of the sum of the two tensions in the belt under consideration.

32 The intersection of this vertical line with that curve in the middle group which is likewise marked $\frac{c^2}{t_0^{2.5}} = 1$, and which accord-

ingly expresses all possible values of the difference of the two tensions, has also been encircled.

33 Taking the readings of the point encircled in the bottom curve, we find

$$\delta_1 = \frac{T_1}{T_0} = 1.81, \text{ and } \delta_2 = \frac{T_1}{T_0} = 0.535$$

We therefore get

$$T_1 = 1.81 \times 175 = 316.75 \text{ lb. and}$$
$$T_2 = 0.535 \times 175 = 93.63 \text{ lb.}$$

34 From this we again get

$$P = T_1 - T_2 = 316.75 - 93.63 - 223.12 \text{ lb.}$$

as the effective pull of the belt. Also, $T_1 + T_2 = 316.75 + 93.63 = 410.38$ lb. as the sum of the tensions, as against $175 \times 2 = 350$ lb., the initial sum.

35 But usually we would not be interested in the separate values of the tensions, and then we would read off directly by the encircled point in the middle group of curves,

$$\frac{p}{t_0} = \frac{P}{T_0} = \frac{T_1 - T_2}{T_0} = 1.275$$

which gives $P = 1.275 \times 175 = 223.12$ lb., the same answer as above.

36 If also interested in the sum of the tensions, we would read this off directly by the point encircled in the top group of curves,

$$\frac{t_1 + t_2}{t_0} = \frac{T_1 + T_2}{T_0} = 2.345$$

which gives $T_1 + T_2 = 2.345 \times 175 = 410.38$lb., the same answer as above.

37 The solution of problems involving long horizontal belts is thus readily enough effected by means of this diagram, but a little consideration will also make it evident that the difference in results obtained by taking the length of a belt into account and by neglecting the same is but slight, except for greater lengths and lower initial tensions than are usually employed in practice. Ordinarily, therefore, we

would use merely the very bottom curves in the bottom and top groups, and the very top curve in the middle group of curves in the middle section of the diagram, which curves are all marked

$$\frac{c^2}{t_0^{2.5}} = 0$$

and thus make no difference between horizontal and vertical belts except for exceedingly long belts.

DESCRIPTION OF DIAGRAMS, FIG. 1, 2, and 3

38 We will now take up the consideration of diagrams Fig. 1, 2, and 3, whi h form the basis of the large diagram in Plate 2. These diagrams are worked out theoretically for vertical belts only, but may be applied without hesitation to horizontal belts of the lengths usually met with in practice.

39 In his paper Notes on Belting, Mr. Taylor showed, as already mentioned, the economy of running belts under much lower tensions than those commonly figured on in proportioning a belt to do a certain amount of work.

40 He also divided the belts with which he dealt into two classes: those whose dimensions he could not well increase over what he found in use, such, for instance, as the cone belts on lathes and other machines provided with a cone pulley in a limited space; and those he could readily increase by providing larger pulleys, such as the belts leading from line-shafts to the counter-shafts of machines.

41 For belts in the first class he adopted higher tensions than for those in the second class. He also devised a set of belt-clamps provided with spring balances, by means of which he could make sure that a belt was put up under a specified initial tension, and could also ascertain its fall in tension at any time desired.

42 All this seems to be the first efforts made by an engineer to pay any systematic attention to the belting in a shop, which even today is usually left entirely to the rule-of-thumb method of the machinist or millwright.

43 The reason why Mr. Taylor had adopted, and accordingly recommended, lower belt-tensions than usually counted on in proportioning a belt to do a certain amount of work, was that a belt quickly loses its tension if it exceeds a certain amount, and thus in order to maintain such a tension, approximately, requires frequent retighten-

ing, which is a source of too much expense and leads to a rapid destruction of the belt. See Fig. 4 and Par. 8.

44 Taking Mr. Taylor's data as a starting point, the writer has finally adopted the rule, as a basis for his use of belts on belt-driven machines, that for the driving belt of a machine the *minimum initial tension* must be such that when the belt is doing the maximum amount of work intended, the *sum of the tension on the tight side of the belt and one-half the tension in the slack side will equal* 240 *lb. per square inch of cross-section for all belt speeds*; and that for a belt driving a countershaft, or any other belt inconvenient to get at for retightening or more readily made of liberal dimensions, this sum will equal *160 lb.*

45 Further, the maximum initial tension, that is, the initial tension under which a belt is to be put up in the first place, and to which it is to be retightened as often as it drops to the minimum, must be such that the sum defined above is *320 lb.* for a machine belt, and *240 lb.* for a counter-shaft belt or a belt similarly circumstanced.

46 The reason for adopting a uniform sum of the tension in the tight side and one-half the tension in the slack side, as mentioned above, instead of either a uniform initial tension, or a uniform maximum tension alone, is that the aim has been to get as uniform periods as possible for the retightening of belts at all speeds.

47 But evidently, while the maximum tension in a belt must be the greatest factor in determining the rapidity with which the belt will lose its tension as a whole, the accompanying tension in the slack strand or side must also have some influence, though not proportionally to the same extent; and hence, the idea occurred to the writer of taking it into account in the manner and to the extent stated.

48 On the diagram Fig. 1, various formulae have been plotted for 240 lb. as the constant sum at all speeds of the tight tension and one-half the slack tension per square inch cross section of belt; for a coefficient of friction that varies with the velocity according to Formula 13 in the Appendix; and an arc of contact of 180 deg. The relations of the various tensions in the belt for all speeds may there be studied to great advantage. It will thus be seen that the centrifugal tension completely balances the initial tension, at a belt speed of 6805 ft. per minute.

49 On the diagram in Fig. 2 some of these formulae have likewise been plotted, with the lesser value of 160 lb. as the constant sum of the tight tension and one-half the slack tension, but for the same values of the coefficient of friction and the arc of contact. Here the

centrifugal tension balances the initial tension at the speed of 5557 ft. per minute.

50 The diagram, Fig. 1, represents the writer's practice in connection with machine belts; that in Fig. 2 his practice in connection with counter-shaft belts (see Par. 44). Both diagrams were used as the basis for the construction of Plate 2, and for the slide rule illustrated in Fig. 5.

51 In the diagram Fig. 3 are given the horse power outputs per square inch of section of belts running under the conditions imposed in the diagrams Fig. 1 and 2.

52 It will be seen that the maximum output is 13.8 h.p. per square inch of a belt under the conditions imposed in Fig. 1, and that this is for a speed of about 4000 ft. (more exactly 4032 ft.); and that for the conditions imposed in Fig. 2, the maximum horse power is 7.46 per square inch, and that this is for a speed of about 3250 ft. (more exactly 3247 ft.)

DESCRIPTION OF DIAGRAM PLATE 2

53 In the diagram, Plate 2, the data given on the diagrams, Fig. 1, 2, and 3, for a belt of one square inch of section, and an arc of contact of 180 deg., have been so modified that almost any problem relating to belting of any size and any arc of contact can be solved.

54 This will best be illustrated by the following two examples:

55 *Example 1.* The maximum cone step on the counter-shaft of a lathe is 22 in. in diameter and wide enough to carry a 3 in. double belt. The speed of the shaft is to be 300 r.p.m. Assuming the thickness of a 3 in. double belt to be $\frac{5}{16}$ in., and the arc of contact of the belt to be 170 deg.: (*a*) What pull can the belt be counted on to exert, and what horse power will it transmit with this pull? (*b*) Under what initial tension will the belt first be put up, and retightened from time to time? (*c*) And what minimum initial tension must it not be allowed to fall below to insure the above-determined pull without undue slip?

56 *Solution.* To get the answer to question (*a*), we first turn to the small bottom portion of the diagram Plate 2, and on its right hand side note the point reading 300 r.p.m. From this we pass horizontally to the left until we intersect the vertical line from the point reading 22 in. on the scale of pulley diameters at the bottom line of the diagram. From the point of intersection we follow the diagonal line upwards to the bottom line of the main portion of the diagram, and there read the velocity of the belt to be about 1700 ft. per min.

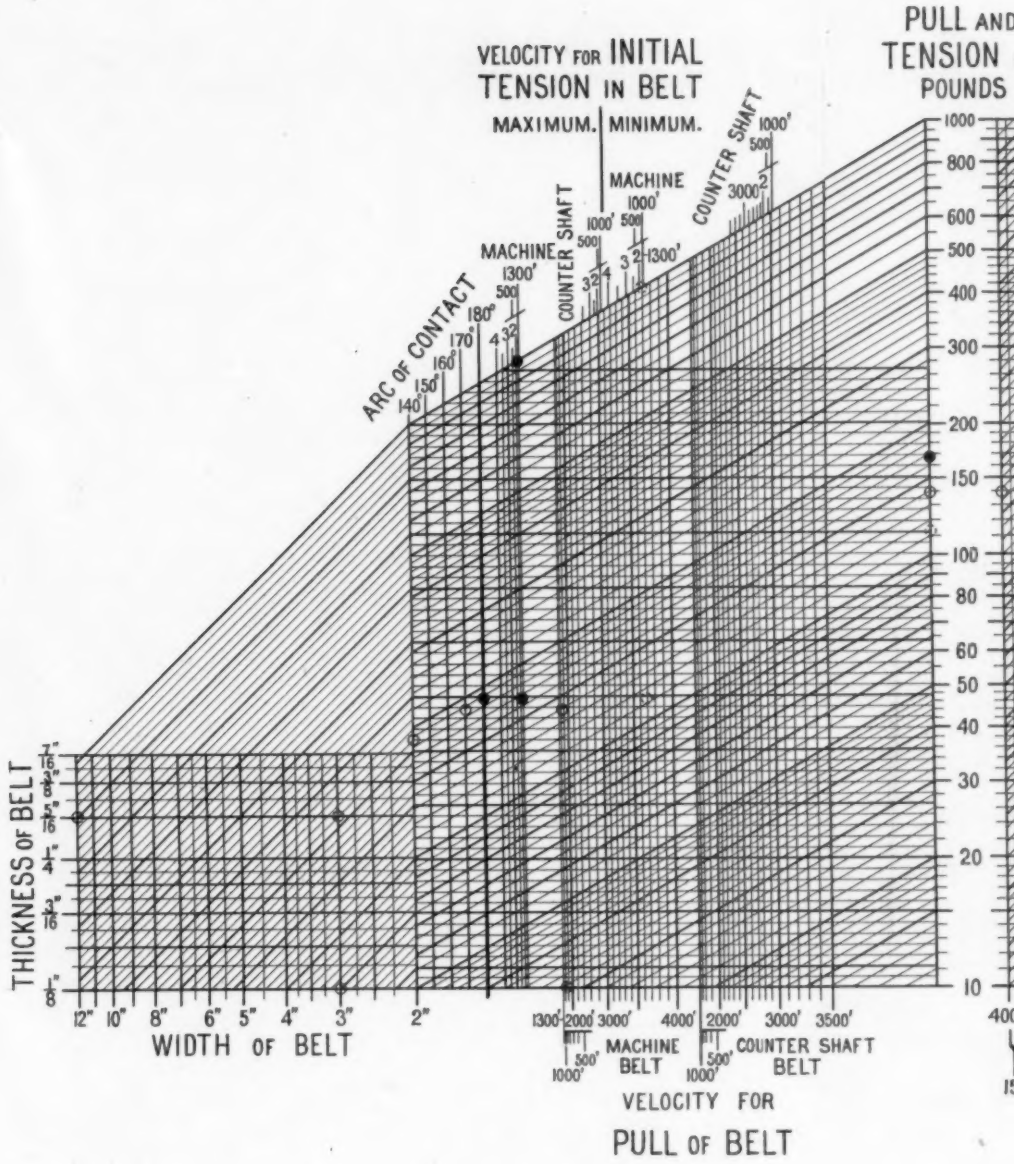
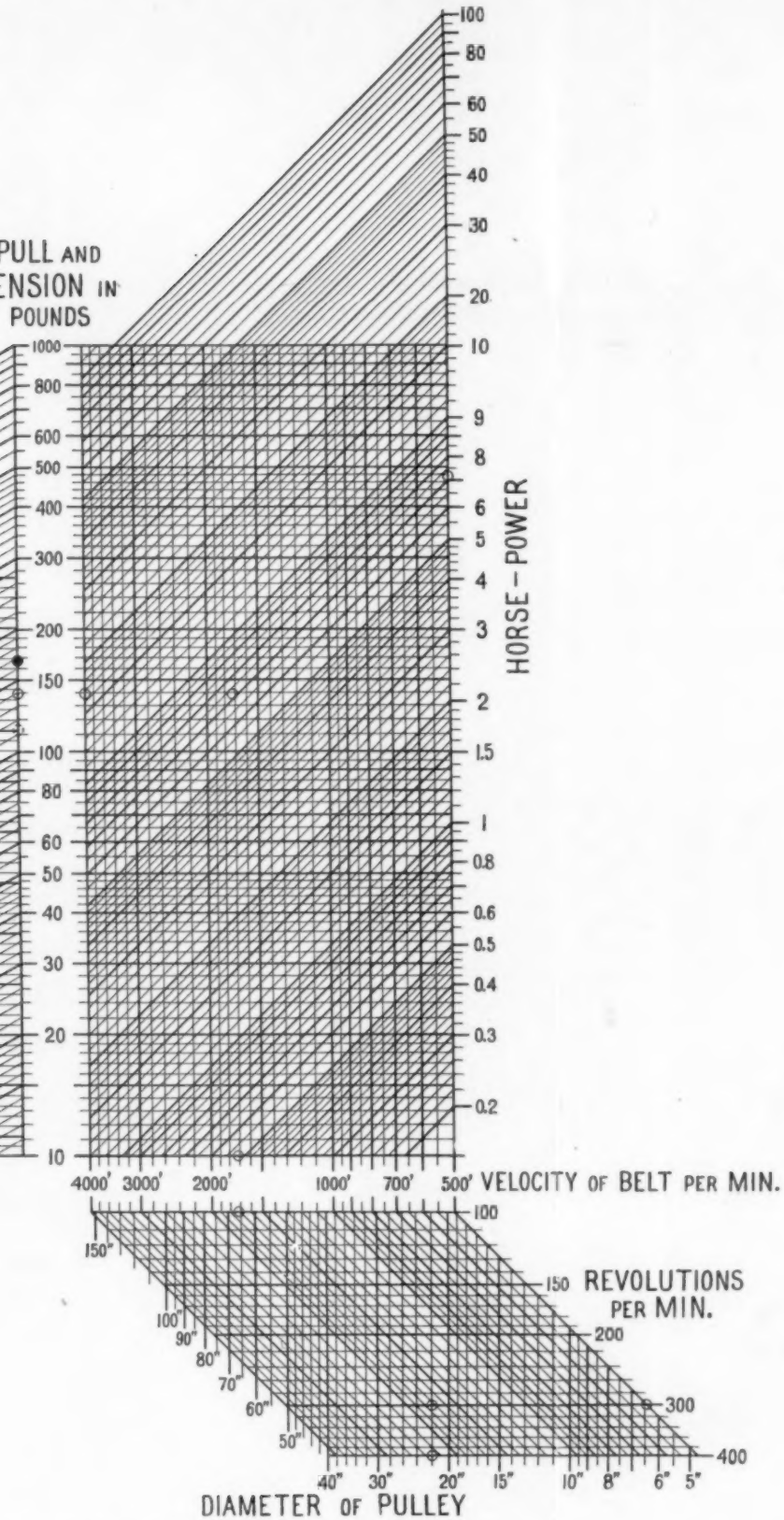


PLATE 2 GENERAL BELTING DIAGRAM INCORPORA



CORPORATING THE AUTHOR'S PRACTICE

FOLDER No. 2

57 We now note the point that corresponds to this belt speed of 1700 ft. per min. in that scale on this same bottom line of the main diagram which is marked "Velocity for Pull of Machine Belt" and interpolate a vertical line extending upwards from this point. Then leaving this for the time being, we turn to the extreme left hand portion of the diagram and there note the point corresponding to the belt thickness $\frac{5}{16}$ in. on the vertical scale to the extreme left, and also the point marked 3 in. on the scale of belt widths at the bottom of the diagram. Following these two points, respectively horizontally to the right and vertically upwards, until they intersect, we now follow the diagonal from this point of intersection until it terminates against the vertical line marked 170 deg. at the top of the diagram, in the field marked "Arc of Contact," and then continue horizontally until we intersect the interpolated vertical line for the belt speed 1700 ft. already noted.

58 From the point of intersection we follow the diagonal until we meet the vertical scale of pounds, on which we now read the belt pull to be 140 lb.; and continuing horizontally until we meet the vertical line extending upwards from the point corresponding to the belt speed originally found on the scale of belt speeds in this section of the diagram, and from this line diagonally to the vertical scale of horse power, we read off the horse power transmitted to be 7.2. All of these movements are indicated on the diagram by little circles around the various points of intersection.

59 To get the answer to question (*b*), we proceed exactly as before, with the width and thickness of the belt, except that we follow the diagonal across the portion of the diagram headed "Arc of Contact" until we meet the border line for 180 deg. From here on we proceed horizontally until we meet the vertical line that corresponds to the belt speed 1700 ft. in the field marked "Velocity for Maximum Initial Tension for Machine Belt." From the point of intersection on this vertical line we then pass diagonally to the scale of pounds, and there read the maximum initial tension to be 168 lb. Those movements for this solution on the diagram that differ from those for the answer to question (*a*), are indicated by little filled in circles around the various points of intersection noted.

60 For the answer to question (*c*), we proceed in every respect as we did for question (*b*), except that we of course proceed from the point corresponding to the belt speed 1700 ft. in that field of the scale on the top line of the diagram which is marked "Velocity for Minimum Initial Tension for Machine Belt." The answer read off on

the vertical scale of pounds is 113 lb. The movements for this solution on the diagram that differ from those for the answer to question (*b*), are indicated by little dotted circles around the points of intersection.

61 *Example 2.* The counter-shaft in Example 1 is to be driven by a belt to run at a speed of 3000 ft. per min. (*a*) What diameter of pulley is required to give this belt speed? (*b*) What pull must the belt transmit? (*c*) What width of double belt must be used? (*d*) And what will be the initial tension under which the belt must be put up, and to which it must be again retightened after falling to the minimum? (*e*) What will be its minimum tension?

62 *Solution.* (*a*) As the counter-shaft is to make 300 rev. and the belt is to run at 3000 ft. per min., we turn to the small diagram at the right hand bottom corner of the main diagram, proceed horizontally to the left from the point marked 300 on the scale of revolutions on the right, until we meet the diagonal line from the point marked 3000 on the horizontal scale of velocities. From the point of intersection we then go vertically down to the scale of pulley diameters, and there read off 38 in. as the nearest even diameter.

63 (*b*) To get the pull of the belt we remember that the cone belt was found in Example 1 to transmit 7.2 h.p. We therefore note that point on the vertical scale of horse powers at the extreme right of the main diagram which corresponds to 7.2, and then follow the diagonal from this point towards the left, until we meet the vertical line extending up from the point marked 3000 on the scale of velocities on the bottom line of this portion of the diagram. From this point of intersection we continue horizontally to the left to the vertical scale of pounds, on which we then read off the pull 80 lb.

64 (*c*) From the point corresponding to these 80 lb. we now continue diagonally to the left until we meet the vertical line extending up from the point corresponding to the belt speed 3000 on the scale marked "Velocity for Pull of Counter-Shaft Belt" at the bottom of this central portion of the main diagram. From this point we continue horizontally to the vertical line corresponding to the arc of contact, which, not being given, we will assume as 160 deg., and then again diagonally in the extreme left hand section of the diagram. Any simultaneous readings of width and thickness from points in the diagonal along which we are now moving, will then give a proper belt, and assuming as in Example 1 a thickness of $\frac{5}{16}$ in., we find the width to be $3\frac{1}{2}$ in.

65 (*d*) To find maximum initial tension for this belt, we proceed

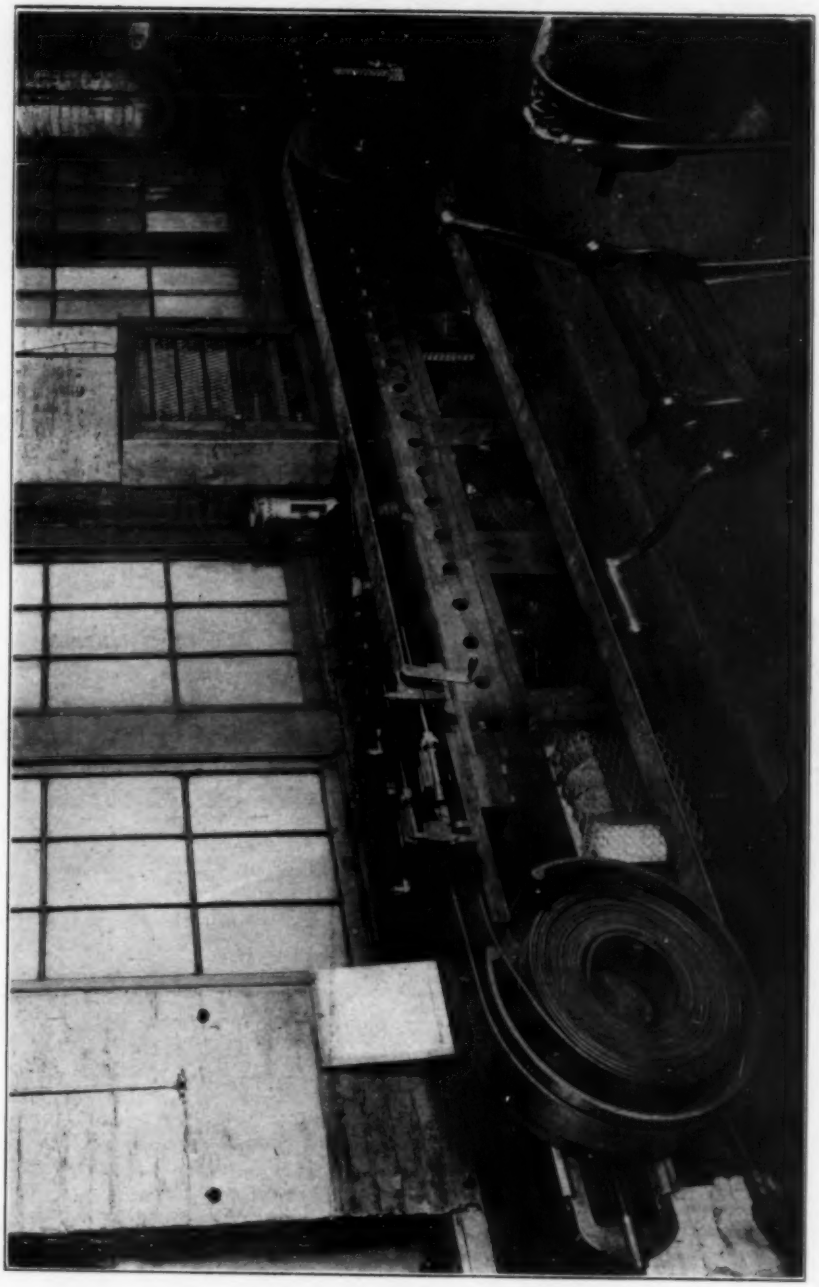


FIG. 6 BELT BENCH IN THE SHOPS OF THE LINK BELT COMPANY, NICETOWN, PHILADELPHIA, PA. THE USE OF THE BELT SCALES IS SHOWN IN DETERMINING THE LENGTH OF A NEW BELT

exactly as in Example 1, except that we use the scale marked "Velocity for Maximum Initial Tension for Counter-Shaft Belt" at the top of the middle section of the diagram, and then read this off on the scale of pounds as 157 lb.

66 (*e*) Similarly, we find the minimum initial tension to be 97.5 lb.

MEANS OF SECURING AND MAINTAINING DEFINITE TENSIONS IN BELTS

67 In his paper, Notes on Belting, Mr. Taylor referred to belt-clamps provided with spring balances for weighing the tension in a belt. In the case of endless belts these scales are put directly on a belt in its final position over its pulleys, while in the case of a belt with wire lacing, this is cut to length under the required tension on the specially designed belt bench illustrated in Fig. 6. As will be seen, this bench is provided with a pair of pulleys which can be so adjusted that a tape-line will measure the same around these pulleys as over the pulleys on which the belt is to run. A belt cut and laced to give a certain tension when the bench pulleys have been properly adjusted, will then be of a length to assume the same tension over its own pulleys.

68 This indirect way of securing a desired tension in a belt was first suggested by our fellow member, Mr. Gullow Gulowsen, who also made the drawings from which the first bench and the first improved belt scale were made by the Bethlehem Steel Company in the year 1900.

APPENDIX[1]

ELASTIC PROPERTIES OF BELTING

The only experiment recorded by Mr. Lewis to establish the elastic properties of leather belting is the following:

2 "A piece of leather belting 1 sq. in. in section and 92 in. long, was found by experiment to elongate $\frac{1}{4}$ in. when the load was increased from 100 to 150 lb., and only $\frac{1}{8}$ in. when the load was increased from 450 to 500 lb. The total elongation from 50 to 500 lb. was $1\frac{1}{16}$ in., but this would vary with the time of suspension, and the measurements here given were taken as soon as possible after applying the loads."

3 These data have been plotted in Fig. 1, in which they are remarkably well represented by the formula

$$L_t = 92\left(1 + \frac{\sqrt{t}}{830}\right)$$

in which L_t is the elongated length of the belt, and t the load or tension per square inch.

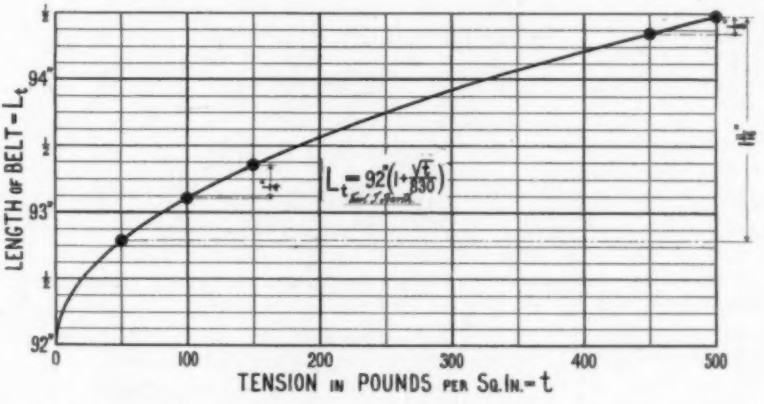


FIG. 1 PLOT OF EXPERIMENTS ON A PIECE OF BELTING 1 SQ. IN. IN SECTION AND 92 IN. LONG. TEST MADE BY WILFRED LEWIS AT THE WORKS OF WM. SELLERS & CO., PHILADELPHIA, IN 1885

[1]The full development of the mathematical formulae of this paper, with some other related matter, is given in an unpublished supplement to the Appendix, which is on file in the Library of the Society for the use of members who wish to verify the mathematical work.

4 On the strength of this formula the writer originally established the theorem that *the sum of the square roots of the tensions in a belt is constant for all loads*, when no attention is paid to the weight of the belt.

5 However, he soon realized that it would not be safe to build a theory on a single experiment of this nature; and hence, in July 1901, while in the employ of the Bethlehem Steel Company, he undertook a series of similar experiments, and obtained permission of William Sellers & Company to use their emery testing machine for that purpose with the assistance of their shop engineer, Mr. Leonard Backstrom.

6 Nine pieces of belting were tested in all. The results upon one of those pieces are shown in Fig. 2, which is typical of all of them. Similar diagrams representing the other tests are filed with the unpublished supplement to the Appendix. In all cases the tests were made as rapidly as the loads could be adjusted and the extensometer readings taken.

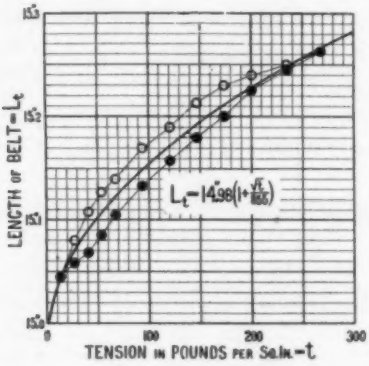


FIG. 2 PLOT OF EXPERIMENTS ON A PIECE OF SLIGHTLY USED DOUBLE BELTING $3\frac{5}{16}$ IN. WIDE BY $\frac{3}{8}$ IN. THICK

Test made by the writer at the works of Wm. Sellers & Co., Philadelphia, in 1901. The Supplement contains eight similar plots.

7 Each piece was several times subjected to a complete cycle of loads between two extremes. During the first few cycles the belts invariably showed different results, but always gave, eventually, practically the same readings for a number of cycles in succession, and these are the readings plotted in the figures.

8 The small filled-in circles represent the readings for an increasing load and the small open circles those obtained for a decreasing load. It is rather astonishing how much lag is shown by every belt. Unquestionably this has some influence on the law of change of tension in a belt, from its minimum to its maximum, along its contact with a pulley. This matter has been given some consideration in the supplement.

9 On account of this lag, apparently it would have been desirable to subject some of these belts to a series of smaller cycles, each between adjacent limits of the load. The best the writer could do with the results obtained was to average

the loop of each cycle by means of a parabolic curve, and thus obtain a value for the constant E for each belt on the supposition that the formula

$$L_t = L\left(1 + \frac{\sqrt{t}}{E}\right) \qquad [1]$$

is approximately correct. In the various formulae given, however, L is not the original 15 in. of length to which the extensometer was originally adjusted for each belt, but an ideal length only, for the estimation of the relations between the tension and the stretch for values never approaching close to zero.

10 But the best experiments for ascertaining the relations between the tension and the stretch in belts are unquestionably those by Prof. W. W. Bird, published in his paper on Belt Creep, read at the Scranton Meeting in 1905.

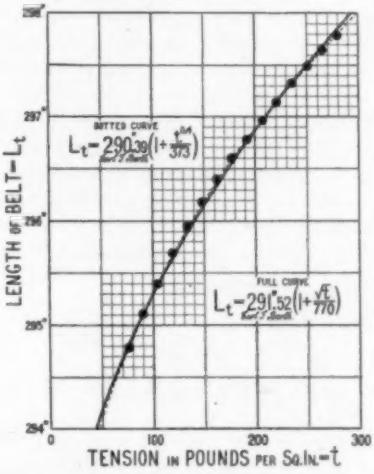


FIG. 3 PLOT OF EXPERIMENTS ON A SINGLE 4-IN., ENDLESS, RUNNING BELT

Test made by Prof. Wm. W. Bird[1] at the Worcester Polytechnic Institute. The Supplement contains a similar plot on a Single 6 in. laced belt. See Professor Bird's paper on Belt Creep in Volume 26 of the Transactions for 1905.

11 These have been replotted by the writer after making some slight corrections in the lengths given by Professor Bird, by allowing for the influence of the sag in the belts, and in the tensions given, by the addition of the estimated centrifugal tension, which was not measured by Professor Bird. The centrifugal tension was estimated after obtaining from Professor Bird the information that the belts were run at a speed of about 1000 ft. per minute. One of the diagrams is shown in Fig. 3 and another has been filed with the supplement.

12 The plots made by the writer differ further from those in the original paper in that he laid off the tension in pounds per square inch of section of the belts.

13 It will be noted that the results have been approximated both by a dotted line representing a special form of the broadly general formula

$$L_t = L\left(1 + \frac{t^n}{E}\right) \qquad [2]$$

and by a full line representing this same formula with the special value $\frac{1}{2}$ for n; or in other words, formula 1. For regularity of results, these experiments are remarkable.

14 While the dotted curves with their more complicated formulae represent the experiments more closely, the full curves with their simpler formulae also cover the results so well that they may be looked upon as an excellent justification for the assumption previously made by the writer on the strength of the experiments made by Lewis and himself, namely, that *within the limits of ordinary working tensions of a belt, the difference between the lengths of a belt at different tensions is proportional to the difference between the square roots of those tensions.*

15 This proportion is implied in the general formula 1, when by L we imply, not necessarily the slack length of a belt, but an ideal slack length on the basis of which the formula gives reliable results between ordinary working limits of t.

16 Taking the average of the values of E in all twelve sets of experiments we get 895. Leaving out two experiments, one with a value of E exceeding 1000 and another for which E was less than 800, and taking the ten remaining experiments with values of E between 800 and 1000, we get 890; while if we take the average of only the two experiments by Professor Bird we get only 861.

17 As will be seen hereafter, the writer has adopted 864 as an average working value, because this figure, combined with certain other constants, results in the simple final constant coefficient 0.04 in the right member of Equation 5. For an average practical working formula on which to build an improved theory for the transmission of power by leather belting, we thus have

$$L_t = L\left(1 + \frac{\sqrt{t}}{864}\right) \qquad [3]$$

in which L_t equals the length of a belt under the unit tension t when its slack length is L.

18 However, it will appear later on, that the value 864 adopted for E has significance only in the formulae developed for long horizontal belts, as E disappears in these formulae when the weight of the belt is neglected.

LAW OF VARIATION IN THE TWO TENSIONS OF A LONG HORIZONTAL BELT

19 In developing an expression to represent the law of variation in the two tensions of a long horizontal belt, the free strands of the belt only are considered, and then, for the sake of argument, are assumed to be attached to the ends of two double levers fulcrumed in the middle, as shown in Fig. 4 and 5.

20 That the parts of the belt in contact with the two pulleys remain at practically constant length independent of any variation in the tensions of the two strands, and thus have no material influence on this variation, will be shown in the Supplement.

21 In Fig. 4 the levers are parallel to each other, and the two strands of belting whose normal slack lengths l are supposed to be equal, must form equal catenaries under equal unit tensions t_0, corresponding to the equal initial tensions in the strands of a belt continuous over its two pulleys.

22 In Fig. 5 the levers have been moved through equal angles in opposite directions, thus tightening the bottom strand to the unit tension t_1, and slackening the top strand to the unit tension t_2, with corresponding changes in the respective catenaries.

23 Under this arrangement the sum of the chords of the catenaries remains constant under all variations of the tensions, a condition that corresponds to that of an actual working belt over its two pulleys, which remain at a constant distance apart under all conditions of tension in the belt

24 In considering the problem, the customary approximations in dealing with catenaries were made, in connection with Formula 1 for the elastic properties of leather belting, and then the following general formula developed for the relations between the unit working tensions t_1 and t_2 in a belt, as dependent on the initial unit tension t_0, the weight W per square inch of cross section of each free strand, and the elasticity constant E in Formula 1.

$$\sqrt{t_1} + \sqrt{t_2} = 2\sqrt{t_0} + \frac{W^2E}{24}\left(\frac{1}{t_1^2} + \frac{1}{t_2^2} - \frac{2}{t_0^2}\right) \qquad [4]$$

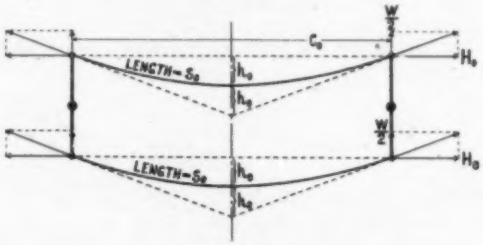


FIG. 4

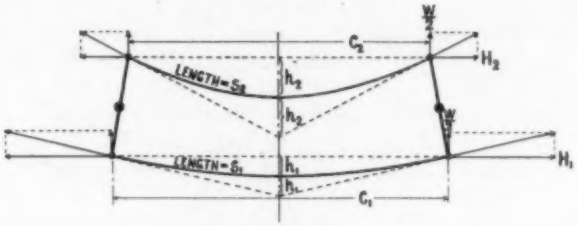


FIG. 5

FIG. 4 AND 5 REPRESENTATION OF CHANGE IN TENSIONS AND SAGS IN STRANDS OF A HORIZONTAL BELT

25 The average weight of a cubic inch of leather belting being about $\frac{1}{30}$ lb. we may write $W = \frac{l}{30}$, in which l is the center distance in inches between the two pulleys of a horizontal drive; and by also assuming 864 as the average value of E, as already done in Formula 3, we may write more specifically

$$\sqrt{t_1} + \sqrt{t_2} = 2\sqrt{t_0} + 0.04l^2\left(\frac{1}{t_1^2} + \frac{1}{t_2^2} - \frac{2}{t_0^2}\right) \qquad [5]$$

26 For very short belts the last terms in Equations 4 and 5, which are only tentatively solvable equations, become very small as compared with the term $2\sqrt{t_0}$, and by neglecting it entirely we write for vertical belts as well as for short horizontal belts, or even approximately for all belts:

$$\sqrt{t_1} + \sqrt{t_2} = 2\sqrt{t_0} \qquad [6]$$

This formula may be enunciated as a new theorem of the relations of the tension in a belt, thus: *Under any variation of the effective pull of a belt, the sum of the square roots of the tensions in the two strands remain constant,* as against the old fallacious supposition that the sum of these tensions remains constant.

27 However, without entirely neglecting the last term in Formulae 4 and 5 above, these may be made solvable with respect to t_1, by first approximating the value of this term by the substitution in it alone of an average relation of the tensions t_1 and t_2. As will appear from a study of the Diagram Plate 1 in the body of the paper, such an average relation of the tensions is

$$t_1 = \frac{t_0^2}{t_2}$$

28 Substituting this in the last term of Equation 5 and then solving this with respect to t_1 we get

$$t_1 = \left[2\sqrt{t_0} - \sqrt{t_2} + 0.04\, l^2 \left(\frac{t_2^2}{t_0^4} + \frac{1}{t_2^2} - \frac{2}{t_0^2} \right) \right]^2, \qquad [7]$$

which gives practically identical results with the original Equation 5 for such values of t_0 and l as fall within ordinary practice.

29 If we express the tensions t_1 and t_2 in terms of the initial tension t_0 by writing

$$\delta_1 = \frac{t_1}{t_0} \text{ and } \delta_2 = \frac{t_2}{t_0}, \text{ Formula 7 reduces to}$$

$$\delta_1 = \left[2 - \sqrt{\delta_2} + 0.04 \frac{l^2}{t_0^{2.5}} \left(\delta_2^2 + \frac{1}{\delta_2^2} - 2 \right) \right]^2 \qquad [8]$$

in which form it is under certain circumstances more readily applied.

TESTING FORMULA 8 BY THE RESULTS OF LEWIS' EXPERIMENTS ON HORIZONTAL BELTS

30 In Fig. 6 the two tensions simultaneously obtained by Mr. Lewis in one series of his experiments have been plotted in terms of the initial tensions of the belt, the tensions in the tight side of the belt being laid off horizontally and the tensions in the slack side vertically, in the same manner as is done on the Diagram Plate 1, in the body of the paper. Similar diagrams, representing five additional series of experiments made by Lewis, are filed with the supplements to the Appendix.

31 However, as the apparatus used by Mr. Lewis did not measure the centrifugal tension in his belts, and as he had no occasion to calculate values of this

quantity, these have been calculated for the present purpose, and added to the effective tensions tabulated by Mr. Lewis.

32 Each experiment is represented by one of the small filled-in circles, and is numbered the same as in the tables from which the experiments were taken from Mr. Lewis' paper.

33 Unfortunately, but very naturally, the initial tension did not remain constant throughout a set of experiments, and in plotting the tensions it was therefore necessary to estimate for each individual experiment where the initial tension was between the values measured at the beginning and at the end of each set of experiments. This was done by assuming that the initial tension measured at the beginning of a set of experiments held good for the first experiment, and that the initial tension measured at the end of a set of experiments held good for the last experiment, and that there was an equal drop for each experiment.

34 That the initial tension was not constant during each set of experiments is the reason why the actual tensions obtained were not plotted, but instead their ratios δ_1 and δ_2 to their respective initial tensions.

35 In each figure, Equation 8 is also given with the center distance of the pulleys for each particular belt introduced as an approximate value of l

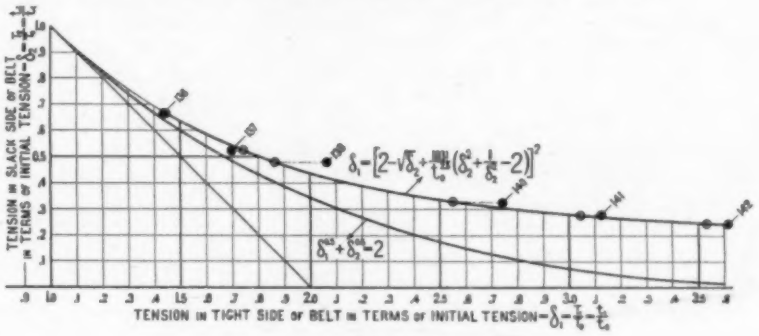

TENSION IN SLACK SIDE OF BELT IN TERMS OF INITIAL TENSION
TENSION IN TIGHT SIDE OF BELT IN TERMS OF INITIAL TENSION
$\delta_1 = \left[2 - \sqrt{\delta_2} + \frac{w l}{t_0 b}\left(\delta_2^2 + \frac{1}{\delta_2} - 2\right)\right]^2$
$\delta_1^{0.5} + \delta_2^{0.5} = 2$
136
137
139
140
141
142

FIG. 6 PLOT OF EXPERIMENTS BY WILFRED LEWIS

Horizontal Double Belt $2\frac{1}{4}$ in. wide, $\frac{5}{16}$ in. thick and 32 ft. long. 20 in. Pulleys. Average Value of Initial Tension $t_0 = 70$ lb. per sq. in.

36 By the introduction of the value obtained for δ_2 in each experiment plotted, a corresponding value was calculated for δ_1 by the formula mentioned, and this value also plotted, and then a curve was drawn to cover the points thus calculated. The points themselves are indicated by the little circles drawn around them.

37 The close coincidence between the curves representing Formula 8 and the experimental results, is certainly all that can be desired in the way of an experimental verification of the soundness of Formulæ 7 and 8.

38 The other, lower curve drawn on each diagram represents the relation between the tensions in a belt when the influence of its weight is neglected, as given by Equation 6, which is also given on each diagram, in the form

$$\delta_1^{0.5} + \delta_2^{0.5} = 2 \qquad [9]$$

BELT CREEP AND ITS INFLUENCE ON THE COEFFICIENT OF FRICTION BETWEEN A BELT AND ITS PULLEY

39 In the paper on his experiments, Mr. Lewis drew the conclusion that the friction between a belt and its pulley varies greatly with its velocity of slip, so that the greater the slip the greater the friction. But as he did not make a substantial study of the elastic properties of leather, upon which the phenomenon of *belt creep* depends, he had no means of distinguishing in his experiments between the necessary slip due to the creep of the belt, and the amount that was slip pure and simple of the belt as a whole.

40 In most of his experiments the belt did an amount of work that called for much greater friction between the belt and its pulley than that corresponding to the creep of the belt alone, and this resulted in additional or true slip that produced the friction needed to make the belt exert the pull called for.

41 By means of Formula 3 it was possible also to derive a formula that gives a good idea of the actual creep of a belt in terms of the tensions in its two strands, which was the object of Professor Bird's paper on Belt Creep, from which the experiments plotted in Fig. 3 of his paper were taken.

42 This formula, the mathematical development of which is given in the Supplement, is

$$v = \tfrac{1}{2}(V_1 - V_2) = \frac{V_1}{2}\,\frac{\sqrt{t_1} - \sqrt{t_2}}{864 + \sqrt{t_1}} \qquad [10]$$

in which

v = actual average velocity of the creep of the belt on each of its two pulleys.

V_1 = velocity of the tight strand of the belt, which is the same as the circumferential velocity of the driving pulley.

V_2 = velocity of the slack strand of the belt, which is the same as the circumferential velocity of the driven pulley.

43 The total creep of the belt on both pulleys together expressed in per cent of V_1 is then

$$x = \frac{100\,(\sqrt{t_1} - \sqrt{t_2})}{864 + \sqrt{t_1}} \qquad [11]$$

44 It must be borne in mind, however, that Formulae 10 and 11 take account of creep only, and have nothing to do with any additional slip due to a sliding of the belt as a whole over its pulleys, though the expression

$$v = \tfrac{1}{2}(V_1 - V_2)$$

taken by itself always represents the total sum of the average creep of the belt and the additional sliding of the belt as a whole, over each of its pulleys, when such additional sliding does take place.

45 Considering the matter in this light Mr. Lewis calculated and tabulated the velocity of sliding from the observed loss in speed between the pulleys in his various experiments.

46 In Tables 1 and 2 appear some of the data thus tabulated by Mr. Lewis. However, instead of tabulating merely the effective tensions measured by him, the centrifugal tensions have here been figured and allowed for, and then the total

tensions thereby obtained subsequently converted into tensions per square inch of cross-section. A column has also been added giving the percentage of average slip due to belt creep alone, as figured by Formula 11.

47 It will be seen that in most of the experiments the velocity of sliding greatly exceeded the average due to the elastic creep alone, and that thus the belt as a whole slid over the pulleys in addition to the elastic creeping, thus showing that the friction corresponding to this creep alone was not enough to produce the pull the belt was called on to perform.

48 The relation between the total average velocity of sliding of the belt on each pulley, and the corresponding coefficient of friction calculated by Mr. Lewis by the formula

$$\text{Ratio of Effective Tensions} = e^{\phi\alpha}.$$

and also copied in Tables 1 and 2 of this paper, is plotted in the diagram Fig. 7 and in a similar diagram of the Supplement. On these diagrams is also shown a curve representing the equation

$$\phi = 0.6 - \frac{2}{4 + v} \qquad [12]$$

in which ϕ is the coefficient of friction and v the total average sliding velocity of the belt in feet per minute. These results were obtained from belts that had been in active service, and tested without the application of any belt dressing.

49 As a somewhat conservative average the curve is seen to cover the results obtained with the belts in a normal condition in a highly satisfactory manner.

50 The question now arises, What coefficient of friction ought to be assumed in calculating the pulling power of a belt at any given speed? In view of the foregoing it does not seem right to assume an average coefficient for all belt speeds Nor would it be right to base it on an average total sliding velocity of a belt corresponding to a fixed percentage of the belt speed, for even a very low percentage would mean a very high sliding velocity in the case of a high-speed belt, while a high percentage would mean only a moderate sliding velocity in the case of a slow-speed belt, and it would seem that the speed with which a belt slides over its pulley would principally determine the life of a belt that meets with no accident.

51 After considerable study over the subject, the writer has assumed a variable coefficient of friction expressed by the formula

$$\phi = 0.54 - \frac{140}{500 + V} \qquad [13]$$

in which V is the velocity of the belt in feet per minute.

52 Equating Formulae 12 and 13 we get

$$v = \frac{160 + 0.88\,V}{85 + 0.03\,V}$$

as the velocity of sliding on each pulley in terms of the velocity of the belt itself.

53 As the percentage of slip between the circumferential speeds of the two pulleys of a belt is twice the percentage of the average total velocity of sliding v of the belt over each pulley, we may now write

$$x = \frac{200\ v}{V} = \frac{200}{V} \cdot \frac{160 + 0.88\,V}{85 + 0.03\,V} \qquad [14]$$

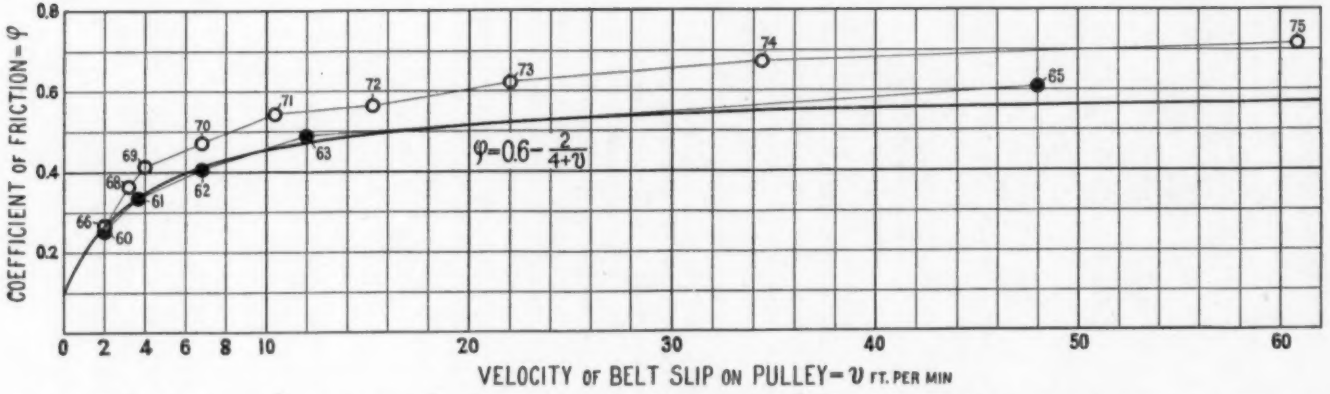


FIG. 7 PLOT OF EXPERIMENTS MADE BY MR. WILFRED LEWIS, AT THE WORKS OF WM. SELLERS & CO., PHILADELPHIA, IN 1885, ON A SINGLE BELT 5½ IN. WIDE BY $\frac{7}{32}$ IN. THICK, BELT IN ORDINARY WORKING CONDITION

Average total initial tension in experiments 60 to 65, $= t_0 = 81.6$ lb. per sq. in.
Average total initial tension in experiments 66 to 75, $= t_0 = 127.5$ lb.

as an expression for the percentage of slip corresponding to the coefficient of friction expressed by Formula 13.

54 In Table 3 are listed simultaneous values of ϕ and x as expressed by Formulae 13 and 14 and also, for the sake of comparison, for ϕ figured by the formula

$$\phi = 0.6 - \frac{400}{800 + V},$$

which is the value of ϕ by Formula 12, for $v = \frac{V}{200}$, that is, for a uniform slip of one per cent at all belt speeds.

55 A study of Fig. 7 and the similar figure in the supplement will show that the variation in the coefficient of friction with the initial tension of the belt is so conflicting as to make it best to leave this out of consideration entirely, and so adhere to the customary assumption that the coefficient of friction is independent of the intensity of the pressure. Therefore, as our whole theory of the variation of the coefficient with the belt speed rests entirely on the formula

$$\text{Ratio of Effective Tensions} = e^{\phi\alpha}$$

this will be used unhesitatingly in spite of the fact that its absolute validity will be disputed in the Supplement for a number of reasons

EFFECT OF CENTRIFUGAL FORCE IN A BELT

56 While the effect of the centrifugal force in a fast-running belt seems to have been fully understood by all who have previously treated the subject before this Society, it is still but imperfectly understood by many engineers and mechanics; and has even been treated in a wrong way by certain text-book writers who have followed the work on belting by the late Professor Ruleaux. It has therefore seemed desirable, in the Supplement to the Appendix, to go into details on this part of the subject.

57 Subsequently, the following general formula is developed for the loss in effective tension in a belt, due to its centrifugal force:

$$t_c = \frac{w}{300\,g} V^2$$

in which

t_c = loss in effective tension per square inch of cross-section of belt.
V = velocity of belt in feet per minute.
w = weight of one cubic inch of belting in pounds.
g = acceleration of gravity in feet per second.

Substituting $w = \frac{1}{3}\frac{3}{0}$, as in paragraph 25, and $g = 32\frac{1}{6}$, we have more specifically

$$t_c = 0.000003454\ V^2, \tag{15}$$

which is substantially the same formula as that given in Mr. Nagle's paper, Formula for the Horse-Power of Leather Belts, read at the Hartford meeting in 1881.

FORMULA FOR PULLING POWER OF A HORIZONTAL BELT IN TERMS OF ITS INITIAL TENSION

58 For a horizontal belt on pulleys of the center distance c and one square inch of cross-section we now have, by substituting c for l in Formula 7,

$$t_1 = \left[2\sqrt{t_0} - \sqrt{t_2} + 0.04\, c^2 \left(\frac{t_2^2}{t_0^4} + \frac{1}{t_2^2} - \frac{2}{t_0^2} \right) \right]^2 \qquad [16]$$

$$\text{Ratio of effective tensions} = \frac{t_1 - t_c}{t_2 - t_c} = e^{\phi\alpha} \qquad [17]$$

$$t_c = 0.000003454\, V^2 \qquad [15]$$

$$\phi = 0.54 - \frac{140}{500 - V} \qquad [13]$$

$$p = t_1 - t_2$$

in which formulae

c = center-distance of pulleys, in inches.
t_0 = initial tension.
t_1 = tension in tight strand or side.
t_2 = tension in slack strand or side.
t_c = centrifugal tension; or, more correctly, loss in effective tension due to centrifugal force.
p = effective pull.

(the last six quantities are per square inch of cross-section of belt.)

e = basis of Naperian System of Logarithms, 2.71828.
ϕ = coefficient of friction between belt and pulley.
α = the lesser arc of contact of belt on pulleys, in radians = $\frac{\pi}{180} \times$ arc in degrees.
V = velocity of belt in feet per minute.

59 However, an attempt to combine these five equations algebraically to obtain an expression of p in terms of t_0, V, and α, leads to an equation solvable by trial only, and for this reason the diagram Plate 1, the use of which was explained in Par. 11 to 24 in the body of the paper, was constructed to effect the solution graphically.

60 For the construction of this diagram Equation 5 was used after substituting c for l as above,

$$\delta_1 = \frac{t_1}{t_0}, \text{ and } \delta_2 = \frac{t_2}{t_0},$$

as in deriving Formula 8 from 7. It thus became

$$\sqrt{\delta_1} + \sqrt[3]{\delta_2} - 2 = 0.04 \frac{c^2}{t_0^{2.5}} \left(\frac{1}{\delta_1^2} + \frac{1}{\delta_2^3} - 2 \right), \qquad [18]$$

and was in that form solved tentatively to obtain a series of points in a series of curves, each representing a certain value of the factor $\frac{c^2}{t_0^{2.5}}$. On the diagram these curves form the bottom field of curves in the middle section.

61 The equation was in each case first solved to obtain an approximate value only of δ_1, in terms of an assumed value of δ_2, by resorting to the approximation δ_2^2 for $\frac{1}{\delta_1^2}$ in the right-hand member of the equation, as shown by Formula 8.

62 Then by substituting this approximate value of δ_1 in the term $\frac{1}{\delta_1^2}$ in the right-hand member of Equation 18 this was again solved for a still closer value of δ_1.

63 For the lesser values of $\frac{c^2}{t_0^{2.5}}$, this closer value of δ_1 differed inappreciably from the first approximation, while for the greatest values plotted on the diagram, the equation was solved twice to get the values actually plotted.

64 However, these greatest values of $\frac{c^2}{t_0^{2.5}}$, never occur in the practical use of belting, and hence the very construction of the diagrams under consideration proved the validity of the approximations δ_2^2 for $\frac{1}{\delta_1^2}$, which is equivalent to the approximation $\frac{t_2^2}{t_0^4}$ for $\frac{1}{t_1^2}$ resorted to in Par. 27 in modifying Equation 5 to Equation 7.

FORMULAE FOR PULLING POWER OF VERTICAL BELTS IN TERMS OF INITIAL TENSION

65 For a vertical belt the relation between the tensions of a belt is expressed by the simple Equation 6, and this can readily be combined with the rest of the equations listed in Par. 58 (in the manner done in the supplement to the Appendix), which leads to the following formula:

$$p = 4\left(\frac{e^{\phi\alpha}+1}{e^{\phi\alpha}-1} - \sqrt{\frac{4\,e^{\phi\alpha}}{(e^{\phi\alpha}-1)^2} + \frac{0.000003454\,V^2}{t_0}}\right)t_0 \qquad [19]$$

66 This formula is simple enough, though a great improvement over the one derived on the erroneous supposition that the sum of the tensions is constant for all loads.

67 By means of this formula the pulling-power of a belt can easily be determined in terms of its initial tension, however, for a uniform unit initial tension for all speeds, the unit tension in the tight side would vary so much that belts running at different speeds but tightened to a uniform maximum unit initial tension, and allowed to run until this had dropped to a uniform minimum tension, would require re-tightening at greatly different periods.

68 As already pointed out the writer has arrived at the conclusion that the periods at which belts running at different speeds will have to be re-tightened, will be nearly constant if they are all made to do their work at such initial tensions as under full load will result in the same *sum* of the *tension* in the *tight side* and

one-half the *tension* in the *slack side* of the *belt*, at the two extremes of the initial tensions, just before and after retightening.

PULLING POWER OF BELTS IN TERMS OF A CONSTANT SUM OF THE TIGHT TENSION AND ONE-HALF THE SLACK TENSION, AT ALL SPEEDS

Vertical Belts

69 This condition is expressed by the equation

$$t_1 + \tfrac{1}{2} t_2 = A = \text{a constant.}$$

Combining this with Equation 17 (in a manner shown in the Supplement) we get

$$p = \frac{(e^{\phi\alpha} - 1)\,(2A - 0.00001036\,V^2)}{2\,e^{\phi\alpha} + 1} \qquad [20]$$

$$t_1 = \frac{2\,A + p}{3} \qquad [21]$$

$$t_2 = \frac{2\,(A - p)}{3} \qquad [22]$$

and

$$t_0 = \frac{4\,A - p + \sqrt{(4\,A - p)^2 - 9\,p^2}}{12} \qquad [23]$$

These formulae are the ones plotted in the diagrams Fig. 1 and 2 in the body of the paper, for $A = 240$ and 160 lb., respectively.

70 By a similar treatment (as shown in the Supplement) we are also able to get an expression for the initial tension in a horizontal belt which gives results of a high degree of accuracy. This expression is

$$t_0 = \tfrac{1}{4}\left[\sqrt{t_1} + \sqrt{t_2} - \left(\frac{c}{5}\,\frac{t_1 - t_2}{t_1\,t_2}\right)^2\right]^2, \qquad [24]$$

which is evaluated by first determining p by Formula 20, and subsequently t_1 and t_2 by Formulae 21 and 22, as for a vertical belt, paragraph 69.

71 However, while this formula is of great theoretical interest, it is hardly of much practical value; as the initial tension determined by it will differ but little from that determined for a vertical belt by Formula 23, except for belts of extraordinary lengths.

72 One very interesting general conclusion may now be drawn for Formula 24; namely, that while actually doing work two horizontal belts of unequal lengths may be under precisely the same tensions, but this being the case, when idle the longer belt will be under a slightly lower initial tension.

73 It appears, however, that the popular notion that horizontal belts drive a great deal more than vertical belts, is not well founded.

TABLE 1

EXPERIMENTS BY WILFRED LEWIS, AT THE WORKS OF WM. SELLERS & CO., PHILADELPHIA, 1885, ON SINGLE BELT 5½ IN. WIDE BY $\frac{7}{32}$ IN. THICK AND IN ORDINARY WORKING CONDITION WITHOUT BELT DRESSING. BELT SPEED = 800 FT. PER MINUTE. THESE EXPERIMENTS ARE PLOTTED IN FIG. 21, WHICH SEE. SEE ALSO HIS PAPER, NO. 198, VOL. 2 OF TRANSACTIONS, TABLE. 1

Experiment No.	Unit Initial Tension t_0	Unit Tension in Tight Side of Belt = t_1	Unit Tension in Slack Side of Belt = t_2	Percentage of Total Slip	Average Velocity of Sliding in feet per minute = v	Coefficient of Friction by the Formula $\frac{t_1 - t_c}{t_2 - t_c} = e^{f\alpha}$	Percentage of Slip due to Belt Creep alone. Formula $s = \frac{100\,(\sqrt{t_1} - \sqrt{t_2})}{864 + \sqrt{t_1}}$
60		125.33	58.67	0.5	2.0	0.251	0.41
61	81.6 lb.	131.42	46.58	0.9	3.6	0.336	0.53
62	per square inch	142.00	42.00	1.7	6.8	0.407	0.62
63		152.41	35.75	3.0	12.0	0.490	0.73
65		179.92	29.92	12.0	48.0	0.610	0.91
66		177.42	77.42	0.5	2.0	0.270	0.52
68		198.25	64.92	0.8	3.2	0.365	0.69
69	127.5 lb.	208.77	58.67	1.0	4.0	0.418	0.77
70	per square inch	219.08	50.75	1.7	6.8	0.472	0.87
71		229.50	46.17	2.6	10.4	0.545	0.95
72		244.08	44.08	3.8	15.2	0.569	1.02
73		256.58	39.92	3.5	22.0	0.623	1.10
74		252.42	35.75	8.6	34.4	0.677	1.13
75		283.66	33.67	15.2	60.8	0.719	1.25

TABLE 2

EXPERIMENTS BY WILFRED LEWIS, AT THE WORKS OF WM. SELLERS & CO., PHILADELPHIA, 1885, ON DOUBLE BELT 2¼ IN. WIDE BY $\frac{5}{16}$ IN. THICK AND IN ORDINARY WORKING CONDITION WITHOUT BELT DRESSING. BELT SPEED 800 = FT. PER MINUTE. THESE EXPERIMENTS ARE PLOTTED IN FIG. 22, WHICH SEE. SEE ALSO HIS PAPER, NO. 198, VOL. 2 OF TRANSACTIONS, TABLE 2

Experiment No.	Unit Initial Tension t_0	Unit Tension in Tight Side of Belt = t_1	Unit Tension in Slack Side of Belt = t_2	Percentage of Total Slip	Average Velocity of Sliding in feet per minute = v	Coefficient of Friction by the Formula $\frac{t_1 - t_0}{t_2 - t_0} = e^{f\alpha}$	Percentage of Slip due to Belt Creep alone by Formula $s = \frac{100\,(\sqrt{t_1} - \sqrt{t_2})}{864 + \sqrt{t_1}}$
105		104.9	47.5	0.3	1.2	0.263	0.38
106	73.5 lb.	123.4	37.5	0.8	3.2	0.395	0.57
107	per square inch	146.0	32.6	1.7	6.8	0.511	0.73
108	171.5	171.5	29.5	4.3	17.2	0.600	0.87
121		403.0	175.9	0.7	2.8	0.267	0.77
124	283.0 lb.	450.0	137.5	1.5	6.0	0.387	1.07
125	per square inch	465.0	124.8	2.3	9.2	0.424	1.17
126		482.2	113.5	3.7	14.8	0.469	1.28
127		497.5	99.2	10.1	40.4	0.523	1.39
128		511.3	227.0	0.5	2.0	0.261	0.85
131	343.5 lb.	557.0	187.2	1.1	4.4	0.350	0.99
133	per square inch	589.5	162.4	1.8	7.2	0.414	1.30
134		603.0	148.2	2.7	10.8	0.450	1.39
135		618.0	134.0	5.1	20.4	0.490	1.49

TABLE 3

RELATIONS BETWEEN COEFFICIENT OF FRICTION, VELOCITY OF SLIDING AND BELT SPEED. SEE PAR. 54.

V = Velocity in feet per minute	Per cent of Slip $= \frac{200}{V} \times \frac{160 + 0.88V}{85 + 0.03V}$	Velocity of Sliding $v = \frac{160 + 0.88V}{85 + 0.03V}$	Coefficient of Friction of $\phi = 0.54 - \frac{140}{500 + V}$	Velocity of Sliding at 1 per cent slip $v = \frac{V}{200}$	Coefficient of Friction of $\phi = 0.6 - \frac{2}{4 + \frac{v}{400}} = 0.6 - \frac{800}{800 + V}$
0	∞	1.88	0.260	0.00	0.100
50	9.432	2.36	0.285	0.25	0.129
100	5.636	2.82	0.307	0.50	0.156
200	3.690	3.69	0.340	1.00	0.200
300	3.010	4.51	0.365	1.50	0.236
400	2.640	5.28	0.384	2.00	0.267
500	2.400	6.00	0.400	2.50	0.292
600	2.227	6.68	0.413	3.00	0.314
700	2.090	7.32	0.423	3.50	0.333
800	1.983	7.93	0.432	4.00	0.350
900	1.889	8.50	0.440	4.50	0.365
1000	1.808	9.04	0.446	5.00	0.378
1200	1.675	10.05	0.458	6.00	0.400
1400	1.566	10.96	0.466	7.00	0.418
1600	1.474	11.79	0.473	8.00	0.433
1800	1.394	12.55	0.479	9.00	0.446
2000	1.325	13.25	0.484	10.00	0.457
2500	1.180	14.75	0.493	12.50	0.479
3000	1.067	16.00	0.500	15.00	0.495
3500	0.974	17.05	0.505	17.50	0.507
4000	0.898	17.95	0.509	20.00	0.517
4500	0.832	18.72	0.512	22.50	0.525
5000	0.768	19.40	0.514	25.00	0.531
5500	0.727	20.00	0.517	27.50	0.536
6000	0.684	20.53	0.519	30.00	0.541
6500	0.646	21.00	0.520	32.50	0.545

A NEW DEPARTURE IN FLEXIBLE STAY-BOLTS

By H. V. Wille, Philadelphia, Pa.

Member of the Society

There is practically no literature on the subject of stay-bolts, and this is particularly true of flexible stay-bolts. The increasing size and pressure of boilers make this subject of vital importance to railroads and to those responsible for the management of that type of boiler in which the firebox is stayed by a large number of bolts.

2 The boiler of the consolidation locomotive, now the prevailing type in freight service, contains about 1000 bolts less than 8 in. long and about 300 of greater length. The large types of Mallet compound locomotives now meeting with much favor have a much larger number, there being 1250 short and 300 long bolts in locomotives recently constructed.

3 In recent years some form of flexible stay-bolt, that is, one having a movable joint, has been very extensively used in the breaking zone of locomotive boilers, but their high cost and the difficulty of applying them, their rigidity from rust and scale, and the fact that their use throws an additional service on the adjacent bolts because of lost motion, has militated against their more general use.

4 It is well known that stay-bolts fail, not because of the tensional loads upon them, but from flexural stresses induced by the vibration resulting from the greater expansion of the firebox sheets than of the outside sheets, but notwithstanding the general acceptance of this theory, engineers have designed stay-bolts solely with respect to the tensional loads. It is quite general practice, it is true, to recess the bolts below the base of the thread, and this has effected a slight reduction in the fiber stress, but practically no effort has been made to design a bolt to meet the flexural stresses or even to calculate their magnitude. This is surprising in view of the simplicity of the calculations to which the ordinary formulae for flexure apply.

To be presented before The American Society of Mechanical Engineers. All papers are subject to revision.

5 Let

F = fiber stress.
E = modulus of elasticity.
I = moment of inertia.
D = diameter.
N = deflection.
L = length.
W = load.

We then have

$$W = \frac{2\,F\,I}{D\,L} \quad \ldots\ldots (1)$$

$$N = \frac{W\,L}{3\,E\,I} \quad \ldots\ldots (2)$$

substituting

$$N = \frac{2\,F\,L^2}{3\,E\,D} \quad \ldots\ldots (3)$$

$$F = \frac{3\,E\,D\,N}{2\,L^2} * \quad \ldots\ldots (4)$$

This formula shows that the stress increases in direct proportion to the diameter and decreases as the square of the distance between the sheets.

6 The application of the formula to service conditions gives the following stresses:

Conditions: Bolt spacing, 4 in. centers.
Assumed expansion, 4/100 in.
Length of bolt, 6 in.

Type	Diameter of Bolt	Flexural Stress
ron	1½ in.	51,500
Iron	1 "	45,000
Iron	¾ "	39,400
Spring steel	1 in. ends $\frac{7}{16}$ in. stem	19,700

7 Iron is universally employed in the manufacture of these bolts and it is not good practice to exceed a fiber stress of 12,000 lb. per

* Testing of Stay-bolt Iron, H. V. Wille, A. S. T. M., vol. 4, 1904.

square inch. It is apparent that stay-bolts in the zone which meets the expansion of the sheets are stressed above the elastic limit and must necessarily fail from fatigue. Fractures always originate at the outside sheet at the point where the bending moment due to the movement of the furnace sheets is greatest.

8 The fractures are in detail, usually starting from the base of a thread and gradually extending inward. Manufacturers of stay-bolt material have endeavored to minimize failures and to meet the unusual conditions of an iron stressed beyond its elastic limit by the supply of specially piled iron arranged with a view to breaking up the extension of the initial fracture. For this reason iron piled with

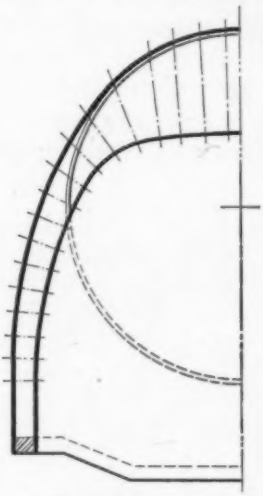


FIG. 1 SECTION OF FIREBOX SHOWING STAY-BOLTS

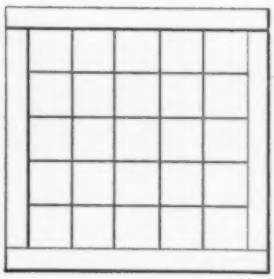


FIG. 2 FAGGOTT PILING FOR IRON FOR STAY-BOLTS

a central section of small bars and an envelop of flat plates has met with much success for this class of service. In a further effort to secure an iron specially adapted to this class of work various forms of shock, vibratory and fatigue tests have been imposed. No design has yet been produced however which permits the employment of material of elastic limit sufficiently high to resist the flexural stresses, although a large class of material particularly adapted to the purpose is available.

9 It is obvious that the remedy does not lie in the use of a slow-breaking material but in the employment of material of sufficiently high elastic limit to meet the conditions of service. It is also possi-

ble to reduce the diameter of the bolt greatly by the use of such a material, thus proportionately reducing the fiber stress in flexure.

10 Stay-bolt material however must possess sufficient ductility to enable the ends to be readily hammered over to make a steam-tight joint and to afford additional security against pulling through the sheets. To meet these conditions the bolt illustrated in Fig. 3 has been designed. The stem is of the same grade of steel as that used in the manufacture of springs. It is oil-tempered and will safely stand a fiber stress of 100,000 lb. per square inch. Its high elastic limit makes it possible to reduce the diameter to $\frac{3}{8}$ or $\frac{7}{16}$ in. or even less. The ends are of soft steel, and it is thus possible to apply and head up the bolt in the usual manner.

11 The employment of a stem of the diameter indicated reduces the fiber stress in flexure to less than one-half that in the ordinary type of bolt and it is of material capable of being stressed to a high degree. It has hitherto been impossible to employ in stay-bolts any of the steels containing chromium, nickel, vanadium or other metaloid

FIG. 3 FLEXIBLE SPRING STEEL STAY BOLT

possessing properties expecially adapted to this class of work, but these steels can readily be used in the stem of the bolt described.

12 The stem of the bolt can be flexibly secured to the end in one of the customary ways, but the flexibility of the bolt does not depend upon a flexible connection. A type of bolt with a relatively inflexible connection, usually one in which the stem screwed into the ends with a running fit, met with the most favorable consideration. Such a bolt is flexible as a spring is flexible, in that it can be deflected to meet the requirements of service without exceeding the elastic limit. In fact the stem may be of a number of pieces, either of plates or small rods, thus increasing its flexibility.

13 The actual breaking strength of the bolt sizes ordinarily employed is shown in the following statement. These bolts were recessed to the base of the thread and tested in the same form as that in which they are employed in service. For comparison the approximate weights of the usual length of bolt are also given. These

weights are for bolts over the entire length, including the squared ends for screwing the bolts into the sheets.

ACTUAL BREAKING STRENGTH OF STAY-BOLTS

Type	Nominal Diameter	Actual Breaking	Weight	Vibrations
Iron	1 in.	32,500	20 oz	6,000
Iron	$\frac{7}{8}$ "	24,500	15 "	5,200
Spring steel stem	1 in. ends $\frac{7}{8}$ in. stem	32,000	10 "	500,000

14 The vibrating test was made by clamping one end of the bolt in a machine and revolving the other end through a radius of $\frac{3}{32}$ in., the specimen being 6 in. long from the end of the right head to the center of the rotating head. A tensional load of 4000 lb. was also applied to the bolts. The best grades of iron bolts break on being subjected to from 5000 to 6000 rotations, whereas the spring steel bolts were

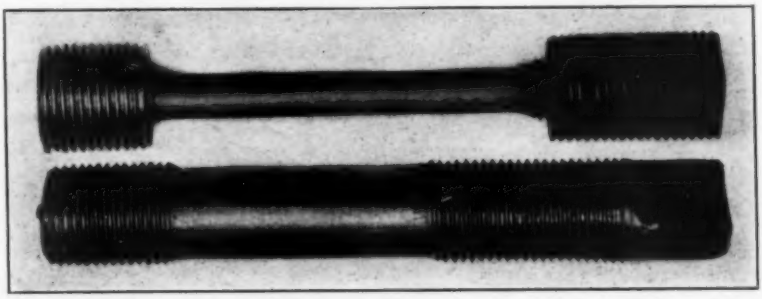


FIG. 4 SPRING FLEXIBLE AND REGULAR IRON BOLTS OF SAME TENSILE STRENGTH.

vibrated 500,000 times without failure, and on some of them the test was continued without failure to 1,000,000 vibrations. These tests demonstrated that the bolt is not stressed beyond the elastic limit under these severe conditions and that the probability of its failure in less severe conditions is very remote.

15 The extent of the expansion which can take place in the firebox of a boiler can readily be calculated.

Distance between stay-bolts, 4 in.
Temperature of inside sheet, 400 deg. fahr.
Temperature of outside sheet, 100 deg. fahr.
Coefficient of expansion, 0.0000066.

Then the expansion between two bolts will equal: $0.0000066 \times (400 - 100) \times 4 = 0.0079$, and each bolt will deflect 0.00395 in. It has been shown that this amount of deflection will stress the usual type of bolt beyond the elastic limit. In practice however one bolt may hold rigidly, throwing the entire deflection on the adjacent bolt, or neither bolt may deflect and the sheet will then buckle. Under this condition the neutral axis will assume the form ABC and the length AB will equal 2.00395 in. and the sheet will buckle to an extent, $BD = \sqrt{2.00395^2 - 2^2} = 0.125$ in. It is obvious that the repetition of a force sufficient to buckle a sheet $\frac{1}{8}$ in. must ultimately lead to a

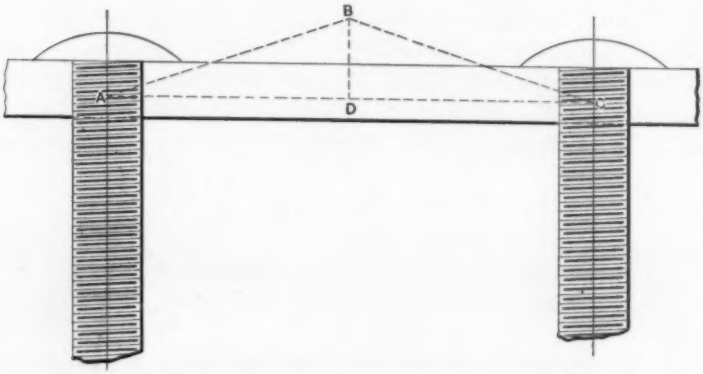


FIG. 5 SHOWING MANNER IN WHICH PLATES BUCKLE WITH RIGID STAYS

crack in the furnace sheets. If, however, the bolt deflects, allowing the sheet to normally expand, the latter will be relieved of these extraneous loads.

16 A bolt of sufficient flexibility to deflect under the forces following expansion, and of material which will not be stressed beyond the elastic limit in resisting these forces, will greatly assist in reducing the cost of boiler maintenance by eliminating broken stay-bolts and reducing the stresses in the furnace plates. If in addition the bolt has a smaller diameter the life of the furnace plates should be further increased, as such a bolt will interpose less obstruction to the circulation of the water in the water legs.

DISCUSSION

THERMAL PROPERTIES OF SUPERHEATED STEAM

By Prof. R. C. H. Heck, Published in May Proceedings.

ABSTRACT OF PAPER

A graphical comparison is made of data from experiments of Knoblauch and Jakob and of Thomas upon specific heat of superheated steam under constant pressure. This shows a marked difference near the point of saturation. The author explains conditions at this point and finds evident errors in interpretation by Knoblauch and Jakob.

Views are given of the critical state of water and a new interpretation is made of Knoblauch and Jakob's results, bringing them into essential agreement with the work of Thomas. The author treats of the higher range of superheat, referring to experiments of Holborn and Henning, and gives a combination of data derived graphically showing the probable manner of variation of specific heat.

There is a table for superheated steam, giving specific heat, heat added above saturation and entropy above saturation. Other data are also referred to, with several comparisons and a statement of the general considerations involved.

DISCUSSION

The Author. The useful end finally sought in a discussion of this subject must be, not the instantaneous specific heat c_p, but the quantity of heat h needed to raise the steam from the saturation temperature to some higher temperature t. Obviously, an exact location (thermally) of the zero or starting-point of this operation is essential. This starting-point of superheat is also the terminus of another most important operation, that of evaporation. Nothing has been made clearer by recent experiment than the fact that the state of dry saturation is exceedingly elusive, and very hard to produce or to maintain in a body of steam.

2 Professor Thomas used in his calorimeter a small current of steam, broken into very thin streams where it passed over the heating surface of the wire coils. While the action observed in the superheating section of the apparatus of Knoblauch and Jakob raises serious doubts as to whether Thomas really secured homogeneous dry steam to which to add his measured superheat, nevertheless the writer is much inclined to believe that Dr. Davis has over-estimated the magnitude of the error involved. It will be noted, however, that the final curves in Fig. 6 of the paper fully embody the idea that Thomas was charging too much energy to the early part of the superheating process.

3 Even if the writer erred in following Thomas in making the curves of c_p on t sweep upward toward saturation, the amount of this error is small. As a concrete case, consider the first line of Table 1, for 1 lb. absolute pressure. In the accompanying tabulation, the first line t_s is degrees of superheat; the second is c, from the table; the third line c' is from an assumed curve which sweeps in to saturation without rise; Δh, in the fourth, is the difference in the heat h,

TABLE 1

t_s	=	0.		10.		20.0		40.0		60.0	80.0
c	=	0.513		0.475		0.465		0.457		0.454	0.453
c'	=	0.450		0.450		0.451		0.451		0.452	0.452
Δh	=	0.44			0.20		0.20		0.08		0.03

over one interval, due to the difference between c and c'. Adding up these values of Δh, we have a total of 0.95, or say one B.t.u. If chargeable to that cause, this would represent the effect of $\frac{1}{10}$ of 1 per cent of entrained moisture. Of course there is uncertainty here, and we need more precise information; but this small possible error in the superheat measured from saturation, is certainly far less than the probable error in Regnault's value of the total heat, measured to that same "point" of saturation.

4 Knoblauch and Jakob made no attempt to go back to zero superheat experimentally. The extension of their curves to this limit is extrapolation, pure and simple, even though made with the best judgment. The writer cannot, however, accept the dictum that c_p must rise to infinity at the critical temperature, no matter what the mathematical deduction on which this idea is based. In the past much mental energy and ingenuity have been wasted in the effort to apply the general mathematical theory of thermodynamics to this

subject of superheated steam, with an entirely insufficient basis of special and exact physical knowledge; nor has any such basis been established up to this time. The relation $c_p = \infty$ belongs to the evaporation process: it means that heat can be imparted under constant pressure without raising temperature. Consideration of the writer's Fig. 3 ought to convince anyone of the impossibility of such action taking place anywhere to the right of the saturation line *ECD*. If the initial c_p need not thus rise to infinity at 689 deg. fahr., it is perfectly permissible to decide that Knoblauch and Jakob may have made their higher curves, which were wholly extrapolated, rise too rapidly toward saturation.

5 It is hard to see any justification for the idea that the lines of superheat h on temperature-rise t_s, which Thomas drew on his Fig. 17, can meet and start at any other point than the origin of the system of coördinates used. What Thomas did was to find, as accurately as he could, the point where any further addition of heat to steam originally slightly moist just began to raise the temperature: then from this point as zero he began to measure the superheat h. Whether he succeeded in getting precise and entirely reliable results is another matter; but curves logically representing such experiments must come to the origin. It is not well to wrench experimental results into conformity with a preconceived rigid assumption, of which the especial merit is the possibility of expressing it by a simple mathematical formula. It would be very difficult to build a physical concept about the idea of some other "origin of superheat" than the zero-point.

6 The presentation of further experimental data in regard to superheated steam will be looked for with great interest: but what we now need is a thorough review of the properties of saturated steam. At present, our knowledge of the heat required above saturation is probably more accurate than what we know about saturated steam.

THE CONVEYING OF MATERIALS

THE BELT CONVEYOR

By C. Kemble Baldwin, Published in June Proceedings

ABSTRACT OF PAPER

This paper has been presented mainly to acquaint engineers who are designing plants with the possibilities of the belt conveyor for handling heavy abrasive materials, and to give them some data which will be of service in the preparation of preliminary designs.

The author believes the following claims for this type of conveyor to be well based:

1 Large capacity with low power requirements.
2 Small maintenance charges.
3 Freedom from breakdown.
4 Light weight of mechanism for large capacities.
5 Material carried comes in contact only with the belt.
6 Perfect alignment is not absolutely necessary.
7 Can be made up into light portable sections.
8 Large overload capacity.

Local conditions have an important influence on the choice of the proper width and character of belt, general arrangement of conveyors, etc., and on these matters the specialist should be consulted.

DISCUSSION

The Author. It is to be regretted that the discussion of this subject has been contributed only by men interested in the manufacture of conveying machinery. When the paper was published the writer sent copies to a dozen large users of belt conveyors asking for comments and such power and cost data as would be of general interest. While many of these users have data which would form the most valuable part of a discussion on the conveying of materials, giving the results of actual experience unbiased by commercial considerations, it is extremely difficult to persuade a works manager to put his notes into shape for publication.

2 Mr. Willcox discusses the use of belt conveyors in salt manufacturing plants. He states that while satisfactory for handling salt from warehouse to vessels in large quantities, they are not satisfactory for handling small quantities from grainers to storage building because of the difficulty of cleaning the return belts.

3 The writer cannot agree with this because he has designed and built within the last six years several very successful plants for handling salt direct from the open pans and vacuum pans. The salt from the open pans contains about 50 per cent hot water, and the main claim for the belt conveyor is, that the salt comes in contact only with the rubber belt, which will not discolor it. There is no difficulty in cleaning the return belts by rotary brushes. After a careful investigation one of the largest salt companies in the country recently made two extensive installations of belt conveyors for handling grainer and vacuum salt. While the reciprocating conveyor described by Mr. Willcox works satisfactorily, it will be noticed that the metal scrapers work in the salt, and must in time drop pieces of rust. It is for this reason, that belt conveyors and not reciprocating conveyors were adopted by the company previously mentioned.

4 In service of this kind we find it best to paint all metal parts of the conveyor with white lead or Atcheson graphite, including the finished surfaces. Rubber belts are used throughout.

5 In the first four paragraphs of his discussion, Mr. Piez makes the following statements:

a That the purpose of the original troughing idlers used with the Robins conveying belt was to *force* the belt to assume the shape of a deep continuous trough.

b That to secure flexibility in the belt, strength was sacrificed.

c That owing to increased loads per unit area of belt due to the deep trough, it was necessary to increase the initial tension largely.

6 From the above he draws the conclusion that, the belt being "defective in structure," the increased "unit strains" brought about a destruction of the bond between the fabric and the rubber coating, which was assisted by the lateral bending due to troughing. The remedy he states was reducing the depth of trough, thus decreasing the load per foot of belt and the unit stress in the belt.

7 The Robins Patent Belt was made with a flexible middle, part of the plies being stopped off at a varying distance from the edges. This formed a belt that would trough of its own weight and had stiff

edges which not only supported the trough between idlers, but gave the desired tensile strength. Being flexible, it was not necessary to "force" it to assume a deep trough, and having extra plies of duck at the edges its strength was not impaired in any way. That the belt is not inherently defective in design is shown by the fact that belts of this type are still running after ten years of continuous service.

8 In Par. 3 Mr. Piez states that the difference in "load per unit area" between a shallow and a deep trough causes such increased tension in the belt that it is destroyed thereby. In Par. 11 he also states that "the no-load readings are frequently from 90 to 95 per cent of the full-load readings."

9 These statements are difficult to reconcile, because if there is only 5 to 10 per cent difference between no-load and full-load, there can hardly be a heavy increase in the tractive force necessary to drive the belt on account of the slightly greater load of a deep trough, over that of a shallow trough.

10 In Par. 5 to 9 inclusive Mr. Piez states:

a That the trough of the idler was reduced because the lateral bending destroyed the belts.

b That the idlers illustrated in Fig. 6 and Fig. 7 of the writer's paper are "shallow-troughed."

c That the tendency is toward the use of an idler that is flat the greater part of its length, the end diameter being 1½ in. larger than the center diameter.

11 The first troughing idlers built by Mr. Robins some 13 years ago were of the three-pulley type illustrated in Fig. 6. The inclined pulleys were about 40 deg. from the horizontal. The angle was soon changed to 25 deg. and 30 deg., depending on the width of the belt and the duty. Mr. Piez calls them "shallow belt conveyors" and evidently agrees with the writer that they are correct. The reduction of the angle was not due to belt troubles, but to the fact that it was soon discovered that there were practical limits to the depth of load that could be put upon a belt, as has been explained by the writer under Speed and Size of Belts.

12 As the deepest practicable load could be obtained by less trough, an angle was chosen that would give: *a*, a reasonable capacity for the width of the belt; *b*, sufficient trough to support the belt between idlers, so that the minimum number of idlers would be required; *c*, enough trough to center the load properly, thus insuring a straight running belt. The angles chosen, namely 25 deg. and 30°

deg., have proved so satisfactory that they have been adopted by most manufacturers.

13 The writer was not aware that there is a tendency to revert to the old type of flat belt—as practically all of the belt conveyors being installed today use idlers with pulleys inclined at 25 deg. or 30 deg. A belt supported on the idler described by Mr. Piez (lower cut Fig. 9 of Mr. Peck's paper), which is flat the greater part of its length, the edges being turned up $\frac{3}{4}$ in., has all the disadvantage of a perfectly flat belt, as a trough of $\frac{3}{4}$ in. would not materially increase its capacity over that of a flat belt. It would not in any way aid in supporting the load between idlers, thus requiring more idlers, and would have no influence on centering the load. Its place, therefore, is in the class of flat belt-conveyors which are practically of the past.

14 In Par. 10, 11 and 12, Mr. Piez states:

a That there is a slip between belt and idler pulleys of the type in Fig. 6 and Fig. 7, due to the retarding action of grease and hub friction.

b That the no-load readings are frequently 90 to 95 per cent of the full-load readings.

15 The writer, in Par. 15, calls attention to the fact that grease lubrication increases the power consumed, but gives good reasons why oil cannot be used.

16 The claim that the idler pulleys quickly become polished on the surface, due to the slip of the belt, is not the case on the conveyors designed and built by the writer. It should be kept in mind that the average belt speed is from 200 to 400 ft. per minute. Therefore the pulley speed is so slow that the weight of the belt alone will turn the pulleys without difficulty. If there is any slip it is not apparent either in the polishing of the pulleys or in damage to either belt or idlers.

17 It is surprising what a small difference there is between the no-load and the full-load readings on the belt conveyors cited by Mr. Piez. It can be explained only by a lack of proper attention to the design and workmanship of the bearings of the idlers and the proportioning of the other parts of the conveyor and its speed to the work to be done—as the writer finds from the results of hundreds of carefully tabulated tests, that the no-load readings vary from 35 to 75 per cent of the full-load readings, depending on the length and speed of the belt and the amount of material handled.

18 The power required to operate a conveyor may be divided into

that consumed by the driving mechanism and that consumed in turning over the idler pulleys. Now, the power required by the former per foot of conveyor does not of course increase proportionately with the length, while the power necessary for turning the idler pulleys varies directly with the length and load of the conveyor. We find in consequence, for short conveyors where the greater part of the power is consumed by the drive, that the difference between no-load and full-load readings is frequently small; whereas for long conveyors, in which the power required by the driving mechanism is a less important factor, the difference may amount to 50 per cent or more.

19 In Par. 62 are given certain percentages to be added to the computed horsepowers required by short conveyors, on account of the predominance of the power consumed by the driving mechanism.

20 Concerning the question of conveyor drives, Mr. Piez says, in Par. 13 to Par. 21, inclusive:

a That the writer places too little value on the use of pulleys of proper diameter.

b That each pulley around which the belt passes under full load should have a diameter of not less than 8 in. for each ply of canvas.

c That the drive shown in Fig. 10 of the writer's paper would be a destructive one to the belt.

d That the drive of a belt conveyor should be located only at the head or discharge end.

e That the multiple pulley drives described by the writer are expedients and not based on sound practice.

21 In Par. 40 the writer cautions against the use of pulleys of small diameters. Table 1 gives "minimum" size of driving pulleys and was so given under the head of drives as a guide to engineers who are designing plants, to enable them to determine clearances.

22 The writer has found it impossible to give a hard-and-fast rule for the diameters of driving pulleys. As Tables 1 and 3 show, it has been found in actual practice that the minimum diameter should be from 4 to 5 in. per ply of canvas, depending entirely on the stress in the belt and on local design conditions. Mr. Piez disregards the fact that Table 1 gives "*minimum* sizes of driving pulleys." Larger pulleys are naturally used when possible.

23 No part of the conveyor design has been given more attention than the drives and a careful tabulation of hundreds of installations

shows that the belts do not fail through separation of the plies due to pulleys of these sizes.

24 Mr. Piez's theory that the pulleys should have a diameter of 8 in. for each ply makes them unnecessarily large, causes very unsatisfactory chutes and, as pointed out in Par. 39, complicates the reduction from motor or engine.

25 Referring to the drive shown in Fig. 10, which Mr. Piez claims would destroy the belt, attention is called to the fact that while the driving pulley transmits motion to the belt, the one at the head merely acts as a bend pulley. The head pulley in such a drive may be smaller than the drive pulley.

26 The drive shown in this photograph has a 48 in. driving pulley, a 30-in. head pulley, with a 9-ply 36-in belt. The best proof that this drive is *not* a destructive one is the record of this conveyor, which has been in operation over three years and has handled over 1,500,000 tons of mine-run coal. The original belt is still in use, and to use the words of the manager of this plant, "It looks good for another million and a half tons."

27 Mr. Piez, whose business has been mainly the manufacture of chain conveyors, would locate the drive of the belt conveyor only at the head end, as would be the case with a chain conveyor. The writer has for the past five years been using the multiple-pulley drive located anywhere in the length of the conveyor, and the results have been most satisfactory. Although hundreds of them are in use, there has not been a single case where the belt separated in the plies, due to the action of the drive. In this connection, accompanying tables will be of interest, both as to the location and the style of drives in three conveying plants designed by the writer.

28 Plant *A* handles mine-run coal on the 36-in. conveyors, capacity 500 to 700 tons per hour. 30-in. conveyors handle fine coal 300 tons per hour. The conveyors have been in operation over three years and have handled over 1,500,000 tons. The original belts are still in use. Fig. 10 is a photograph of drive of Conveyor No. 2. Since these conveyors were started three years ago, every chain conveyor in the plant, except one, has been replaced by a belt conveyor.

29 Plant *B* handles mine-run and fine coal, capacity 500 to 700 tons per hour. Conveyors have been operated about three years and have handled about 1,000,000 tons.

30 Plant *C* has just been started. After making careful inspections of Plants *A* and *B* and investigating all other methods of handling coal, the owners contracted for this plant.

TABLE I LOCATION AND STYLE OF DRIVES, PLANT A

No.	Width Inches	Length Feet	Inclination Degrees	Location of Drive	Type of Drive
1	36	600		Head End	Multiple Pulley
2	36	300		" "	Like Fig. 10
3	36	500	Level, and 20	Center	Multiple Pulley*
4	36	50	20	Tail End	Single " †
5	36	500		" "	Multiple Pulley
6	30	200	20	Center	" "
7	30	400	20, then Level	Head End	Single "
8	30	100		Tail "	" "

* Lifts Material 90 ft.,
† Driven from head of No. 3.

TABLE 2 LOCATION AND STYLE OF DRIVES, PLANT B

No.	Width Inches	Length Feet	Inclination Degrees	Location of Drive	Type of Drive
1	36	300		Head End	Multiple Pulley
2	36	775		Head "	" "
3	36	411		Head "	" "
4	36	473	20, and Level	Center	" "
5	36	673	20, and Level	Head End	" "
6	36	661		Near One End Reversible	" " Single "
7	36	82	20	Head End	
8	36	196	20	Tail "	Multiple "
9	40	100		Head "	Single "
10	36	300	20	Tail "	Multiple "
11	28	177	20	Head "	Single "
12	36	120	20	Center	Multiple "

TABLE 3 LOCATION AND STYLE OF DRIVES, PLANT C

No.	Width Inches	Length Feet	Inclination Degrees	Location of Drive	Type of Drive
1	36	75		Head End	Single Pulley
2	36	315	20	Center	Multiple "
3	36	30	20	Tail End	Single " *
4	30	32		Center	" " †
5	30	50		Head End	"
6	30	50		Head "	" "
7	30	400	20	Center	Multiple "
8	30	400	20	"	" "
9	36	640	20, and Level	"	" "
10	36	55	20	Head End	Single "
11	36	55	20	Tail "	" "
12	36	480		Center	Multiple " †
13	36	180	20, and Level	Tail End	" "

* Driven from Head of No. 2.
† Reversible Conveyor.

31 The writer believes that the foregoing is the best proof that the drive may be located at any point in the length of the conveyor, and that the multiple pulley drive, far from being an expedient, is based on sound practice and good engineering.

32 In Par. 22 to Par. 25 Mr. Piez discusses the writer's power formula and gives two examples. The results obtained from the power data presented by the writer apply to the belt conveyors manufactured by the Robins New Conveyor Company and the claim is not made that these figures will hold good on conveyors of different design.

33 In Par. 22 the writer laid stress on two fundamental points of economical conveyor design:

a The belt should run no faster than is necessary to carry the desired load. The speed may be determined from Table 2.

b The power required should always be figured for the maximum capacity at the chosen speed.

34 If these points are followed there will be a fixed relation between load and speed: therefore T in the writer's formula (load in tons per hour) is a function of both load and speed. While C is a constant for each width of conveyor, T is a variable depending on the desired load, so that if these were combined as suggested by Mr. Piez a large range of values for T would have to be used, resulting in a cumbersome table of doubtful value.

35 Mr. Peck's table of horse power, to which Mr. Piez refers, and the examples given by him in Par. 23, all indicate that they have left out of consideration the relation between speed and load.

36 Take example *A*, 30-in. conveyor handling coal, 253 ft. centers, capacity 215 tons per hour, speed 600 ft. per minute. Assuming coal to weigh 50 lb. per cubic foot, we find on referring to Table 2 that a 30-in. conveyor at 100 ft. will carry 145 tons of material weighing 100 lb. per cubic foot or 72½ tons of coal; to carry 215 tons therefore the speed should be about 300 ft. per minute instead of 600 ft. Naturally the lower speed will require less power and produce less wear and tear on machinery and belt. According to the writer's Formula 9, 18 h.p. will be required, but this is based on a belt-speed of 300 ft. per minute, and not 600 ft., which is bad practice.

37 Take his example *B*, a 30-in conveyor, 386 ft. center to center, speed 348 ft. per minute, carrying 60 tons of coke per hour. When handling such friable material as coke, the belt speed should be as low as possible to avoid breakage. The speed also should be no greater than is necessary to carry the load.

38 Taking coke at 25 lb. per cubic foot, 60 tons per hour, the capacity desired is about 4800 cu. ft. From Table 2 we find that a 30-in. conveyor has a capacity of 2900 cu. ft. at 100 ft. per minute, therefore the proper speed is about 165 ft. per minute instead of 348 ft. According to the writer's formula, this gives 4.25 h.p., 10 per cent being added for gears, and this would drive the conveyor as built by the writer. The excessive powers shown by Mr. Piez's tests are due in part to the excessive speed.

39 Mr. Messiter presents in Par. 10 of his discussion a formula in which a load factor is multiplied by the load in tons per hour and a speed factor by the speed in feet per minute. With this formula it is possible to compute the power required using a high speed and small load. Should such a conveyor at any time receive the full load at the high speed it would result in failure of belt or drive. In the writer's formula, in which the speed and load factors are combined, and where T is a function of both load and speed, this possible error is avoided.

40 Mr. Bennett, in Par. 5 of his discussion, takes exceptions to the writer's statement regarding the waterproof qualities of rubber belts. The rubber friction is applied to the duck in a plastic state. As its consistency is that of putty, it can do no more than coat the duck without reaching its inner fibers. Therefore when affected by a large quantity of water the belts are destroyed, as described by the writer in Par. 27. On the gold dredges the service is extremely severe and the belts must be built to stand excessive abrasion due to large quantities of wet rock. Therefore it is a duty which requires not only waterproof qualities, but also abrasion-resisting properties. The uncovered Balata Belt mentioned by Mr. Bennett should never have been used in this place as it was not provided with any cover to stand the wear. It should have had a heavy rubber cover.

41 Mr. Bennett states in Par. 9 that the heat of vulcanization does not affect the cotton duck of rubber belts. Actual tests, however, show that there is a loss of from 10 to 15 per cent in tensile strength due to the heat of vulcanization.

42 Mr. Messiter, in Par. 2 of his discussion, takes exceptions to the writer's formula for tension in the belt. As he states, the formula gives only the driving tension and neglects the initial tension, due to the take-ups. The substitution he proposes, while correct, is too cumbersome, and that given by the writer gives values that seem close enough for practical purposes.

43 In this connection, the writer would call attention to his

reason for presenting his paper, stated in Par. 74, "To give the engineers who were designing plants some data which will be of service in the preparation of preliminary designs." This purpose made it seem advisable to simplify and make as practical as possible all of the data presented, and to omit all unnecessary detail.

44 As this discussion has brought out some differences of opinion regarding principles and methods employed, it might be considered pertinent to add a word relative to the connection of the writer with the development of the belt conveyor industry, that members of the Society may better judge of the value of the data presented.

45 In the early days of Mr. Robins' activity he had associated with him James Barnes Humphrey, a young engineer of exceptional brilliancy, who personally designed, with no precedent to guide him, not only the various parts of the belt conveyor (many of which remain unchanged today), but also several large and difficult conveyor plants. These installations, although unique and daring in design, were so successful that the future of the belt conveyor as an industrial factor was immediately assured.

46 By his withdrawal from the work in 1900, followed shortly thereafter by his death, the industry can be said to have suffered the loss of a real genius. The writer, who was associated with Mr. Humphrey, and who carried on his pioneer work, desires that full credit be given the mind that rendered such invaluable services in the evolution of the belt conveyor.

47 The data presented represent the results of the writer's observation of thousands of conveyors, as Chief Engineer of a company manufacturing practically only belt conveyors, and all statistics and figures have been proven by actual practice.

BITUMINOUS PRODUCER PLANTS

By Elbert A. Harvey, Published in the Journal for October

ABSTRACT OF PAPER

The Industrial Gas Plant furnishes clean producer gas from bituminous coal. The plant consists of updraft producers from which the gas is made, all three principles of producer gas-making being employed; viz, semi-combustion of carbon, decomposition of steam and volatilization of hydrocarbons; coolers, in which the gas is cooled by water shower; rotary gas washers, by which the tar, dust and other impurities are beaten out of the gas; dry scrubbers, in which the gas is dried; auxiliary steam boilers and engines for operating the gas washers and furnishing steam to the producers, and tar equipment by which the tar from the gas is collected, stored and burned under the auxiliary boiler.

The gas generated by this apparatus is rich, containing all the light hydrocarbons of the coal. It averages 120 B.t.u. per cubic foot from poor coal up to 170–180 B.t.u. per cubic foot from good gas coal. The process has a net average efficiency of 75 per cent. Of the waste 25 per cent, approximately one-half, is in the form of tar which is used in carrying on the gas-making process.

The first installation of this type of apparatus was made in Indiana in 1906, a power plant of 1800 horse power in three units. Average load, 1320 b.h.p. at the switchboard, 24 hours per day, 6½ days per week. Operates on Indiana bituminous coal, consumption being 1.06 lb. per b.h.p. and 1.75 lb. per kilowatt-hour at the switchboard. Total cost for power at this plant, including coal, water, supplies, labor, and investment expenses, on the entire power plant, is .83 cents per kilowatt-hour.

A second installation of this type of apparatus is a 2000-h. p. plant installed in Indiana, furnishing gas to tempering, annealing and other small furnaces in a file works. Coal used is Indiana run of mine, and the total cost of gas, including same items as above, is 24¼ cents for the equivalent of 1000 ft. natural gas.

Another recent installation, a 750 h.p. plant operating on Ohio bituminous coal, furnishes gas for an average power load of 175-kw. with total coal consumption of 2.81 lb. per kilowatt-hour at the switchboard, and a total cost of 1.85 cents per kilowatt. The plant also furnishes gas for furnaces to the amount of the equivalent of 4000 ft. natural gas per hour, total cost being 57½ cents.

This type of plant is completely successful and can be counted on for continuous operation, 24-hour service. It generates a clean gas from bituminous coal, lignite, or other fuels high in hydrocarbon, and is suited for installations of 500 h.p. and above.

DISCUSSION

G. M. S. Tait I would like to ask Mr. Harvey a few questions as to efficiency obtained on the two-cycle engines referred to in his paper. He says they operate in the neighborhood of 10,000 B.t.u. per brake horse power. This type of engine has a great many advantages, but is not recognized as an economical engine on producer gas, and as this is the first instance of its kind in which such a high economy has been claimed on producer gas, the writer would like to know more about it.

2 I am not criticising the type of engine, but merely asking information, as it seems to me that the method of calculating heat unit consumption is subject to considerable error. For example, the author states that the gas is taken off the regular mains and that it was a fuel gas installation. How could the gas used by the engine be identified from the rest of the consumption, if a common main was used?

3 One more question is whether the history of the plant showed any trouble from premature ignitions, my experience being that this type of engine is designed to operate on a gas of very uniform composition, such as natural gas; if there is any way of operating it successfully on a gas containing a varying amount of hydrogen, this information would be of benefit to others.

John C. Parker. The speaker is sure that he is not at variance with the opinion of the members of the Society when he states that the ultimate interests of the gas producer, like those of any other article or commodity entering into commercial competition, are best conserved by recognizing its limitations, correcting them where possible, and where this can not be done, by abstaining from the attempt to enter unfavorable fields. That the gas engine cannot

compete with other forms of prime mover in certain fields is recognized, and in view of this the writer begs to present the following considerations.

2 The comparison between plants in Par. 8 of Mr. Harvey's paper is likely to be misleading if the exact local conditions are not understood. The equipment consists, for instance, of 1950 h.p. in gas engines carrying an average load of 1320 h.p. for 24 hours, which establishes a consumption of 0.95 lb. per b.h.p. per hour, whereas the steam plant carrying essentially the same average load has a consumption of 5 lb. per b.h.p. per hour. The relation between the average load carried and the plant equipment indicates a load factor very near unity for the gas engine plant. If the same conditions do not obtain for steam engine plants comparison becomes unfavorable to the latter. Assuming on the other hand that the same conditions do obtain, so large a coal consumption seems surprising. As the figures for gas producer plants are exclusive of generator steam and power for the operation of the washers it seems fair to eliminate from the charges against the steam engine plant all auxiliaries. From the size of the steam engine plant one infers that a compound condensing plant is used, and a consumption of 5 lb. per *brake* horse power per hour seems very large, even if inclusive of auxiliaries, contrary in fact to the average results obtained at St. Louis by the Geological Survey, and given in Bulletins 316 and 325. The former bulletin shows an average ratio between coal as fired per b.h.p. in the boiler and coal as fired per b.h.p. per hour in the producer, of 2.7, a maximum of 3.7, and a minimum of 1.8; with lignites and semi-bituminous coal the ratios are 2.7, 2.9, and 2.2. In view of this, one speculates as to why in the Indiana Glass Manufacturing Establishment a ratio of 5.27 should obtain.

3 Par. 9 indicates a direction in which improvement is to be expected. Contrast 1.75 lb. per kilowatt-hour at the switchboard with 1.34 lb. per kilowatt-hour at the switchboard, the difference representing the coal consumed by steaming and operating rotary washers. This excess of 30.5 per cent evidently suggests a fruitful field for development. It is to be hoped that the elimination of a necessarily inefficient type of steam engine from the washer and the application of some heat units at present wasted to the production of steam may materially reduce these losses.

4 Referring now to Par. 11 the writer would like to question the labor charges, and the advisability of operating generating plants with 12-hour shifts, especially in important industrial establishments

maintaining continuous activities. As the plant has been in operation during a period of industrial depression it is perhaps entirely possible to secure competent men at the wages mentioned. The speaker is not familiar with industrial and wage conditions in the section of the country in point, but believes that it would be impossible in the State of New York, during periods of normal business activity, to secure a head gas man at not much above 20 cents per hour, even for power gas generation, and indeed doubts whether it would be advisable even if possible. Assuredly 12 cents per hour for the producer man, 25 for the chief engine man, and 15 for the assistant engine man, are too low. In central New York we pay 20 to 25 cents per hour for the crudest kind of common labor working 8 hours daily, and appreciably larger figures than those mentioned for engine men. On comparatively small engines in our community $18 per week is an average wage for a combination engine man and stoker, and more skilled men draw more proportionately.

5 This brings up the whole question of the advisability of employing inefficient labor for this class of work. Since the coal and water bills alone come to approximately $50 daily, and the labor charge is less than one-half of this, it is obvious that the slight saving in the employment of inferior labor may readily more than compensate for itself in wastage. Gas production *is* quite technical, and the proper adjustment—or refraining from indiscriminate adjustment—of the gas engine is of extreme importance. In 1906 the U. S. Geological Survey, Bulletin 316, reporting on this very point, considered a great deal of the undeserved disrepute of gas producer plants due to the inability of the operators.

6 Par. 11 and Par. 12 contain an apparent discrepancy in the matter of first costs. Par. 11 quotes a total investment of $100 per horse power for the plant, which seems reasonable; but the drop to $55 per horse power of equipment seems almost beyond accounting for by the increased size of the plant estimated in Par. 12, although it is stated that 90 per cent of the costs involved is based on actual quotations. The writer would inquire whether these are quotations for the plant installed and whether they are firm offers. The general experience seems to be that estimated costs, based on other than firm quotations in place, are likely to be largely exceeded, and $55 per horse power seems so close as to suggest that some of the installation and housing charges have been overlooked. Surely a large part of this $55 would be covered by comparatively slow-speed electric-generating apparatus, housing and engines, to say nothing

of the producer equipment. In this connection it is interesting to note that the proportionate cost of the Garford Company plant assessed against the power equipment is only $65.52 per horse power of capacity, as against the $100 estimated against a plant of 3.6 times the size. Moreover, it is interesting to inquire whether the plant capacity in the various cases has made provision for spares and for the necessary overload capacity. The writer infers that this is not the case.

7 The tabulated data of the Hydro-Electric Power Commission of the Province of Ontario for 1906 show that the machinery cost decreased from $70 per horse power for 500-h.p. to $65 per horse power for 1000-h.p. plants; while the total cost decreased from $82 in the former to $73 in the latter case. We are evidently, in these sizes, getting to the flat part of the curve, and should not look for a material reduction of the pre-estimated cost in larger installations. Indeed these installations are much larger than the general run of plants. In fact, of the industrial establishments in the United States developing power in 1905, 108.8 was the average horse power per establishment, which, according to the Hydro-Electric Commission figures—for Canada, where the costs, it is true, might naturally be somewhat higher—gives a plant cost of about $100 per horse power as against approximately $70 estimated for steam plants.

8 As to the proper percentage to be charged on the fixed costs, it is practically impossible to borrow or to invest at less than 6 per cent per annum for interest, allowing 5 per cent actual interest and 1 per cent for the cost of floating bonds and organization. The average tax rate for 1905 on actual property valuation in cities of the United States of over 30,000 was 1.71, fixing this figure as a portion of the annual charge which must be borne by any plant. Three per cent for repairs, which is not perhaps inordinately high, together with insurance, would bring this portion of the fixed cost well above 10 per cent. While with a proper sinking fund, 5 per cent would liquidate the plant investment within the reasonable life of the plant, it seems altogether too small a sum, with the ordinary systems of finance, to cover mere physical decay beyond the range where repairs are economical, and even at that would leave nothing to provide for supervision and extraordinary emergencies. It is also to be noted that in all kinds of cases a period of construction and development of load, though less in the case of industrial enterprises than of public service utilities, must be calculated and paid for, which of necessity increases the rate of fixed charges. In view of this

it is a question whether 15 per cent per annum is very nearly adequate. The writer believes that this figure should be increased by at least 5 per cent, and possibly by more, though the exact figure is of course a matter of independent determination for each case.

9 The importance of squarely facing this element in power plant costs is illustrated by reference to any of the author's examples. Referring for instance to Par. 11, the operating charges per day are only $78.70, the fixed charges—and the writer believes these should be much increased—$90. Industrial establishments having a load factor as high as 60 per cent during a 10 or 12-hour working day are extremely rare, the maximum capacity of the plant being carried not more than 6 hours per day on the average. In such a case the fixed element in the cost would become four times as great per horse-power-hour delivered as in the instance given in the article; and as practically none of the costs other than the fuel item would be reduced, because of the necessity of maintaining a labor charge only slightly below the all-day charges if some slight loads are to be carried at night; while the efficiency of course becomes reduced, this would give a daily operating cost reduced to only $140.60 as against a power delivery at the switchboard of 5346 kw-hr. In other words the cost per kilowatt-hour increases from 0.8 to 2.63 cents.

10 As another illustration of the importance of knowing the first costs accurately and estimating the percentage fixed charge, Par. 16 and 46 may be compared. The latter paragraph shows a cost ratio for producer gas almost strictly in the ratio of the coal cost when compared with the former estimate of $24.25 per thousand feet of natural gas equivalent. As the fuel charge in the former case is but a little over one-third of the total cost, manifestly this can not be correct for identical load factors, in which case the $21.50 fuel item would be reduced to $13.45 by the reduction in cost per ton from $2.00 to $1.25. This would make a total cost of $40.50, or approximately 20.22 cents per thousand feet of natural gas equivalent. If, on the other hand, the plant were to be operated as a 24-hour plant, the investment charges would be reduced to approximately one-half, say $8.75 daily. An extra labor charge of $2.50 would presumably be required to take care of the additional night help, considering also all the possible diminution of the day help. This would make a total daily cost of $34.20, or approximately 17.1 cents per thousand feet of natural gas equivalent. It is manifest that in case of the 10-hour operation the fuel cost has influenced by only 4 cents the cost of the product rendered, whereas in the case of all-day operation the economy is as much as 7 cents.

11 These two citations perhaps serve to mark one of the serious limitations of the gas engine, and to point the force of the communication from the Chairman of the Meetings Committee with reference to the wide discrepancy in weights, and presumably in costs, between different types of engine equipments. If the gas producer plant is to meet at all the competition of steam turbine plants, which have the advantages of extremely low first cost and an excellent sustained efficiency over a wide range of load, it is essential that costs should be reduced as speedily as possible to a competitive working basis.

12 Against the excellent low load efficiency of the turbine plant is to be set the modest stand-by losses of the gas producer itself. The author quotes in the case of the Central Indiana plant a stand-by of ½ ton, against an operating consumption of 8½ tons. The ordinary steam boiler furnace is especially deficient in this respect. On the other hand it is interesting to make inquiry into the possibility of rapidly getting gas producer equipments into operation when carried, as is not uncommon in present steam practice as relays and peak plants, in conjunction with water power developments or where heat engines of some type must be used in any case to handle a widely fluctuating load.

J. R. Bibbins. The bituminous producer involves difficulties in the use of various kinds of fuel, which have only been overcome by years of experiment, and I think we have not by any means reached the ideal.

2 Considering the matter fairly, is it essential to make tar-laden gas? Are there any benefits accruing therefrom that would not be realized from a tar-free gas? I must disagree with Mr. Harvey's opinion that a more useful gas may be produced by this method than by a system which involves tar destruction in the producer.

3 There are a number of features in producer gas work which are, to my mind, essential to the ideal system:

a We must have continuous operation—24 hours a day, and 365 days a year, if desired. This has already been achieved through the use of the water seal. And it may be that future development of large power gas plants will demand a continuous system for removing ash from beneath the water seal by some form of conveyor. This is by no means an impossibility. Fuel is already handled entirely by well known mechanical means.

b The fire bed should be open and readily accessible for occasional working, and this must be capable of accomplishment at any time during operation, as otherwise 24-hour service would not be possible. This is done with the open top producer.

c The fire bed must be operated at a sufficiently low temperature to avoid the formation of clinker. This has already been accomplished and is greatly facilitated by the use of a continuous blast laden with sufficient vapor to induce the necessary endothermic reaction.

d Tar and volatiles should be completely converted into fixed gas within the producer. The difficulties and embarrassments surrounding the rapid accumulation of tar are facts which may not be treated lightly, and have undoubtedly retarded to a considerable degree the progress of bituminous work. Tar destruction is an accomplished fact.

e Auxiliary plant should be reduced to its smallest possible dimensions. This covers the incidental by-product process, auxiliary boilers, bulky power-consuming and cleaning apparatus, and large gas holders.

f A moderate heat value is desired, not excessively high, but high enough to permit obtaining a fair rating from the engine, and especially the use of high compression.

4 Here I take issue with the author's statements concerning tar-free gas. Producer practice has conformed in general to a heat value around 125 B.t.u. effective power value; but I am free to say that this standard is seldom maintained in practice, a value of 110 to 115 B.t.u. representing more nearly the actual conditions. This, however, is not so serious a matter as it seems. The decrease in engine rating for gases ranging from 125 to 110 B.t.u. is barely $\frac{1}{3}$ that from 125 to 80 B.t.u.—blast furnace gas. And in fact the loss of power with the lower producer gas could scarcely be found in the actual plant, where other matters, such as valve setting, time of ignition, temperature of intake, jacket temperature, etc., control to so much greater degree the power and efficiency of the engine. This is especially the case in very "high" gas, in which the excess heat value is largely traceable to the higher hydrocarbons. These excessively rich gases are exceedingly difficult to handle in a gas engine owing to the continual variation in their proportions, the necessity of frequent change of mixture, and their tendency to cause premature or delayed combustion.

5 In producer gas, the possibility of high compression is far more important than the addition of a few heat units to the gas. This has recently been demonstrated by some very interesting experiments by Prof. Hopkinson of Birmingham University, which clearly showed that the efficiency of a given gas engine was actually higher with a lean than with a rich gas; i. e., as the richness of the mixture increased, the efficiency decreased. This is borne out in practice by the fact that fully as high efficiencies seem to be obtainable from blast furnace gas as from natural gas with 10 to 12 times the heat value. Prof. Hopkinson attributes this to the increase in the specific heat of the products of combustion taking place at the higher temperatures. It seems true, therefore, that the excessively rich producer gas is not worth striving for, if other more important advantages are lost.

6 That the various points enumerated above are not impossibilities, is demonstrated by a bituminous producer which has been under my observation for some time. This producer has been under fire continuously for over six months without the fire being drawn. It has been run on various kinds of coal, including lignites, slack and run-of-mine, and at various rates of output, from a month's run at standby to a similar period at maximum overload. The producer has made no tar and no clinker. At all loads, and without auxiliary apparatus, it has supplied its own vapor. A gas of moderate heat value has been made sufficiently uniform to operate an engine day after day without changing the mixture or the use of a gas holder. The efficiency of the producer is reasonably good and entirely comparable with that obtainable from a producer making a tar-laden gas. Efficiency and operating runs were made through monthly periods, not for 12 or for 24 hours only. In fact, the commercial value of the gas produced has been determined by a total run of over 1000 hours by an engine under brake load test.

7 With these facts in view, I feel that the crux of the producer gas problem does not lie in the making of an exceedingly rich gas, but rather in conforming to practical standards which are at present demanded by power users and which in the end represent the ideal to which we must all conform.

8 Commenting upon the paper, Mr. Harvey encourages the inference (Par. 41) that an engine may be designed for and operated on either natural or producer gas without any interruption or adjustment. He should have qualified this statement to some degree, as it is evidently quite impossible to develop a rational and efficient

design of gas engine that permits of instant change from an excessively rich to a poor gas. Not only must the port areas be proportioned to the volumes of gas passing through them, but different-sized combustion chambers are necessary to secure the proper degree of compression for the gas to be used. And with port areas sufficiently large to accommodate both gases, it is certain that difficulties in governing would be encountered unless some adjustments were made, owing to the entirely different ranges of valve movement required.

9 In his statement of producer performance, Par. 11, Mr. Harvey submits figures which indicate a rate of combustion in the producer of 50 lb. per square foot of fuel bed per hour. Assuming four producers in operation, this would be equivalent to 25 lb. per square foot per hour. The higher rate is evidently in error: even the lower rate being extraordinarily high, and apparently almost impossible of maintenance in the service contemplated in Mr. Harvey's estimate.

10 In Par. 8 the author mentions No. 8¾ lump as designating a certain grade of Pittsburg coal. Here is an opportunity for the Society to assist in bringing about definite size standards for anthracite and bituminous coal. In my discussion of Mr. Junge's paper on low grade coals, in 1906, I attempted to show the chaotic conditions existing in the size of anthracite fuels, and no doubt bituminous sizing standards are susceptible of equal improvement.

11 In estimating fixed charges (Par. 11) I have of late come to the conclusion that straight depreciation should not be charged against a power property, but rather that depreciation be considered in the form of a sinking fund or annuity. This is simply in conformity with the actual disposition of funds in commercial accounting. Not including the factor of obsolescence, if a plant has 20 years' life, the straight depreciation is 5 per cent, but if applied in the form of a sinking fund at 5 per cent interest, the annuity is 3 per cent. The general tendency is to charge excessive rates of depreciation against power gas properties. I believe, however, that this subject should be considered more in detail and always in accord with actual accounting. The funds written off on depreciation accounts are never allowed to lie idle. They are always invested in some form, and should therefore be regarded in the nature of a sinking fund.

12 The heat value of CH_4 (Par. 15) is given at 919 B.t.u. I believe this is higher than the accepted value. It illustrates the necessity of the Society's adopting at the earliest possible moment a standard table of heat value. Recently I had occasion to compile the values from some half-dozen authoritative sources, none of

which agreed, not even the computations from the original Thomsen experiments.

13 Mr. Harvey considers frequent charging a necessity for uniform gas (Par. 20). Recent extensive experiments on this very point show the opposite results, viz: that an increase in the rate of charging of from 60 to 10-minute intervals had practically no discernible effect on the range of fluctuation in the gas. This means much less labor on the producer. The fluctuation in volatiles is, on the other hand, almost entirely due to disturbance of a partially-coked fuel bed by poking or charging, so that nothing is to be gained by more frequent agitation of the bed.

14 An important point noted in Par. 24 is the relative rate of combustion possible with very rich and very poor coals; i. e., is it possible with a fuel high in ash and low in heating value to fire at a sufficiently higher rate per square foot per hour to compensate for the lower heat value and maintain the same producer output? In other words, is it necessary to reduce producer ratings on low-grade coal?

15 It is interesting to note how nearly the figure for dust content 0.015 gr. per cubic foot, is approximated in actual practice. Tests recently made from a bituminous gas plant, which has been in operation for two years, giving good results and no trouble whatever from dirt, show 0.043 gr. at the beginning of the week and 0.15 gr. at the end. This is from six to ten times the quantity of foreign matter that Mr. Harvey mentions as a limit. I would not suggest a lower standard for impurities in gas, but it is worthy of note that the modern horizontal double-acting type of engine, such as is considered in the report of the Norton test last year, is able to operate under conditions very far from those originally stipulated. The blast furnace plant at Bessemer (near Pittsburg) actually cleans the gas to 0.02 gr. per cubic foot, yet the air at times exceeds the gas in impurities carried. Furthermore, the 500-h.p. unit originally installed at Bessemer for experimental purposes actually operated for 96 hours on crude gas direct from the furnace flues. At the end of the run, an examination of valves, cylinders, packing and pistons showed conclusively, by the absence of any scoring or sticking of parts, that the modern system of mechanical lubrication is able to cope with serious difficulties. From each valve and combustion chamber were removed about two quarts (3.62 lb.) of solid matter, consisting of dust, cinder, coke and limestone, varying in size from dust to $\frac{1}{4}$ in. or more. The problem of cleaning is important, yet the fact remains that many plants are

operating satisfactorily today on gas containing far greater impurities than expected.

THE AUTHOR Mr. Tait asks for further information in reference to the efficiency of a two-cycle engine. His question refers to three 600-h.p. two-cycle Koerting engines operating in Central Indiana on producer gas.

2 In answer I would say that I cannot furnish further information. The statistics in regard to fuel consumption indicate that these engines are actually operating on 10,000 B.t.u. per brake horse power hour. The engines are unquestionably not of the best type, as is evinced by the fact that their manufacturers have withdrawn them from the market and substituted a much better engine. I understand the makers are now putting out engines consuming around 8000 B.t.u.

3 Mr. Parker refers to figures for the two power plants in central Indiana, where it is stated that a gas power plant carrying an average power load of 1320 b.h.p. uses .95 lb. of coal per b.h.p., whereas a steam power plant at the same factory carrying an average load of 1250 b.h.p. uses 5 lb. of coal per b.h.p. It will be seen that I further brought out that the gas plant uses 1.34 lb. per b.h.p., considering auxiliaries, so that the ratio of coal consumption of the two plants is 1 to 3.8 in favor of the gas plant. Mr. Parker brings out the point that the average ratio as shown by government tests is 2.7, and I agree with him that this figure more nearly represents average conditions than the one which I state. I simply gave statistics of one particular instance, representing two actual equipments operating side by side at the same factory.

4 Mr. Parker questions whether the costs for labor given in Par. 11 are not too low. They undoubtedly are lower than average figures throughout the country. They are perhaps lower than they should be. They are, however, accurate, I believe, for the instance in question, and represent what is actually paid at that particular power plant.

5 He also raises the question as to why there should be such wide variation in the cost per horse power in gas power plants, ranging in the figures I gave from $55.00 per horse power to $100.00 per horse power. There is a very wide range in cost between small and large power plants and the decrease per horse power for the bituminous plants is very rapid, up to about 5000 horse power, and beyond that figure the cost per horse power is approximately constant. Further

the 20,000 h.p. plant referred to where the cost is given as $55.00 per horse power was an auxiliary power plant in connection with a water power plant. It was therefore rated at its full maximum and had no reserve or spare unit, as the plant itself was a spare unit. Under ordinary circumstances this equipment would be rated at perhaps one-third less. Further, the cost varies with the fuel used,—the poorer the fuel the more expensive and more extensive the equipment required. In general I would say that the cost of a bituminous producer equipment is such that it will not be used extensively in sizes below 500 h.p. excepting in localities where the use of anthracite coal is out of the question. Under usual conditions for power plants under 500 h.p., the anthracite suction producer presents a better proposition than the bituminous producer, as it is much simpler and less expensive.

6 Mr. Bibbins draws the issue between the processes of generating producer gas by one of the other of which the tar is burned from the gas in the producer and by which the tar is removed by mechanical means. The term "tar-laden gas" does not apply unless it be to gas supposedly burned free of tar, but in reality, on account of faulty apparatus or operation, carrying tar. When gas is cleaned by mechanical process and really cleaned, it is "tar-free," but it carries that other rich hydrocarbon which is not condensable, viz: CH_4.

7 What I claim is this: that when it is attempted to burn the tar from the gas during the generating process there are three drawbacks which have retarded the use of producer gas from bituminous coal. These are as follows:

a In order to burn the tar from the gas continuously without clogging the fuel bed or the apparatus and at the same time actually clean the gas, it is necessary to use two generators and alternate or reverse the draft in the generators. This involves too great first cost; hot operating parts, hard to keep in repair; too much skill for operation.

b With only one generator the process is too critical for ordinary operating conditions. While it may be carried on successfully under expert supervision and undoubtedly has been in several instances, it is always subject to failure in case of clogging of apparatus or failure to produce clean gas.

c Gas cleaned by burning out the tar averages lower in B.t.u. per cu. ft. than when the tar is removed by mechanical

means, and the lighter non-condensible hydrocarbons left in the gas. The average figures range from 80 to 110 B.t.u. for the tar burning process, and 125 to 160 B.t.u. for the process where the tar is mechanically removed. For power purposes the lower value gas may give as good efficiency as the higher, but it has less capacity and larger apparatus is required than with the higher value gas. It should be emphasized, however, that the use of producer gas is not confined to the generation of power. The gas can be used for all lines of industrial work in large and small heating furnaces, and the field for producer equipment in this direction is probably larger than in connection with power plant equipment. At the present time producer gas is used for many heating operations, the principal drawback being that the gas has not sufficient flame temperature for welding and melting and other similar high heats except where regeneration is employed. But in this respect the possibilities of gas averaging 140 heat units are far greater than the possibilities of gas averaging 100 heat units, and I take the position that every development which increases the heat value of producer gas is a decided step in advance.

8 I further claim that the updraft bituminous gas producer, which generates a gas tar-laden only until the tar is removed mechanically, is a practical plant, easily operated continuously by laboring men, and that it is not likely to be clogged or shut down in case operation is not properly carried on; in other words, the plant meets ordinary conditions. If it is efficiently operated it gives the best of results. If it is operated poorly, it gives comparatively poor results, but inefficient operation does not result in complete failure or shutdown. In this respect the updraft bituminous producer is comparable with a boiler plant, and it is hardly an exaggeration to say that this apparatus deserves to be characterized as practically "foolproof."

9 Mr. Bibbins calls attention to the serious drawback of the production of tar without adequate means of handling it or disposing of it. His point in this respect is well taken. Anyone familiar with a plant where tar of this character has been allowed to accumulate without provision for its handling and disposition will recognize the importance of the subject. If it were not possible to provide adequate means, this would prove a serious objection, possibly strong enough to condemn this type of apparatus.

10 However, the handling and utilization of this tar is not a difficult problem. The tar at ordinary temperatures is a stiff, gummy substance almost impossible to handle, but if slightly heated it becomes liquid and can be disposed of by apparatus similar to fuel oil apparatus. This apparatus not only provides for the disposition of the tar, but for utilizing it in generating power for the operation of the plant. It must be recalled that it is only within the last two or three years that such apparatus has been produced and there are today only a few operating plants. While these are completely successful, they do not embody in every respect all the economies which will ultimately become possible. However, progress enough has been made safely to predict that sufficient steam can be generated with this tar for gas making, for producing power for moving the gas, and the heat necessary for handling the tar itself, in which case the tar would come very near providing for the entire auxiliary plant.

11 I agree with Mr. Bibbins that operating an engine on producer gas is a very different affair from operating on natural gas. However, I have called attention to the fact that five 100-h.p. gas engines operating at Elyria, Ohio, are piped both for natural and producer gas and they operate on either gas without change in compression at the option of the engineer. The producer gas pipes are larger than those for natural gas and when a change is made from one gas to the other the valves are adjusted, but this does not interrupt the operation or cause fluctuations at the voltmeter.

12 The point is important simply because in many cases where engines are operating on natural gas the gas is expensive or the supply is giving out and the question arises as to whether producer gas can be substituted.

13 In reference to Par. 11, I do not see how Mr. Bibbins arrives at the conclusion that there was a coal consumption of 50 lb. per sq. ft. per hour. In the plant in Indiana carrying an average load of 1320 h.p., there are five gas generators connected in one battery with a total generating area of 250 ft. They usually operate three of these, employing a total area of 150 sq. ft. The figures given show a coal consumption amounting to 1760 lb. per hr. indicating a consumption per square foot per hour of a little less than 12 lb. This I would say is a normal condition for a bituminous plant operating at two-thirds to three-quarters load. As a matter of fact I would consider 25 lb. per sq. ft. as a maximum consumption, and 15 lb. per sq. ft. as normal for producers of this type.

THE SURGE TANK IN WATER POWER PLANTS

By R. D. Johnson, Published in June Proceedings

ABSTRACT OF PAPER

This paper treats of the momentum of flowing water in long pressure pipes for the supply of hydraulic turbines or impulse wheels and of the control of rate of flow by modifying the momentum in such manner as to obviate harmful effect upon speed regulation of the water-wheels, *without waste of water through relief valves, deflecting nozzles or by-passing it,* as has usually been thought unavoidable where pipes are long, velocity great and pressures high.

This is accomplished by a surge tank near the down stream end of the pressure pipe, atmospheric or under compressed air.

The paper treats of the size of surge tank needed under various conditions of velocity, size and length of the moving water column, and presents a novel device by which the diameter (or area) of the tank may be reduced about one-half, thereby lessening its cost and at the same time improving the pressure regulation for speed control. This device is called the "Differential Regulator" and it may work open to the air or under compressed air where head is high and support lacking. The design of the regulator is fully set forth in this mathematical relation to the balance of the plant.

DISCUSSION

Chester W. Larner[1] Reverting, in conclusion, to Mr. Johnson's equation [7] it has been shown that this equation without modification gives values of y_{max} too small, and when modified by

[1] Continuation of discussion in The Journal for October, which also contains other discussions of this paper. It should here be noted that the symbol V_{max}, which was adopted from the author's paper, is not strictly accurate in the sense in which it is used. It is used to represent the maximum velocity during the first part of the cycle up to the time when demand and supply become equal. It is not the actual maximum penstock velocity, however, as this occurs later in the cycle, V'_{max} is the correct term but both are synonymous as used.

the substitution of V_{max} (or V'_{max}) for V_2, as was done by him in Par. 87, it gives values too large. Obviously there must in every case be some intermediate value of V' which if substituted for V_2 will give the true value of y_{max}. Since when V'_{max} is used the overrun is due to the variation of V', which the equation does not consider, it follows that this value V'' will lie between V'_1 and V'_{max}. In mathematical terms

$$V'' = V'_1 + K\,(V'_{max} - V'_1) \qquad [3]$$

where K is a coefficient less than unity. Since K cannot be determined from theoretical considerations it only remains to be seen if it can be determined with a fair degree of accuracy by any empirical law. If so we should obtain greater accuracy in the use of Equation [7], which would then become

$$y^2_{max} = \frac{RL}{g}\,(V'' - V_1)^2 + c^2\,(V''^2 - V_1^2)^2 \qquad [7']$$

115 Fifteen problems covering a wide variation in the values of R, L, c, and percentage of load increment have been solved by arithmetical integration and the results, together with the assumed conditions, are given in Table 1. Values of V'' were obtained by substituting the values of y_{max} found by arithmetical integration in Equation [7'] and solving for V''. Values of K were obtained by substituting

TABLE 1 RESULTS BY ARITHMETICAL INTEGRATION AND CORRESPONDING VALUES OF K

	Assumed Conditions						By Arithmetical Integration at Intervals of one sec.					
Ex	h	R	L	V_1	Load Change Per ct.	c	V'_1	V'_{max}	y_{max}	V''	K	$\frac{RL}{c}$
1	150	0.1	2000	11	20	0.1	13.20	14.18	9.56	13.72	0.53	2000
2	250	0.1	2000	12	15	0.2	13.80	14.78	14.84	14.34	.55	2000
3	300	0.2	2000	12	20	0.05	14.40	14.95	10.78	14.85	.82	8000
4	400	0.2	3000	13	15	0.1	14.95	15.42	11.62	15.25	.64	6000
5	500	0.2	6000	14	10	0.15	15.40	15.82	12.61	15.66	.62	8000
6	300	0.25	4000	12.5	15	0.1	14.38	15.21	15.64	15.02	.77	10000
7	500	0.25	10000	13.5	10	0.15	14.85	15.43	17.69	15.31	.79	16667
8	300	0.3	4000	12	15	0.05	13.80	14.55	15.10	14.42	.82	24000
9	500	0.3	10000	13	10	0.1	14.30	14.84	17.60	14.75	.83	30000
10	150	0.1	4000	13	10	0.1	14.30	15.33	9.00	15.00	.68	4000
11	500	0.3	5000	12	13	0.15	13.56	13.99	14.74	13.88	.74	10000
12	100	0.3	2000	13	13	0.05	14.69	19.63	22.95	18.00	.67	12000
13	200	0.227	5000	13	6	0.1	13.82	14.51	8.80	14.35	.77	11300
14	200	0.227	5000	12.5	15.5	0.1	14.44	16.47	22.77	15.95	.74	11300
15	200	0.227	5000	13	12	0.1	14.54	16.13	18.07	15.73	.75	11300

the values of V'' thus found in equation [3]. V'_1 is the initial value of V' and is, of course, V_1 increased to correspond to the additional load.

116 The values of K vary considerably, and after investigation it was found that K is a fairly consistent function of $\frac{R\,L}{c}$, the values of which are given in the last column of Table 1. When these points were plotted on coördinate paper and a rough curve drawn through them, the curve bore a general resemblance to the upper left-hand quadrant of an elipse with major axis horizontal. The relation between K and $\frac{R\,L}{c}$ was therefore assumed to be of the form

$$Ax^2 + By^2 + Dx + E\,y + F = 0$$

where

$$x = \frac{RL}{1000\,c} \text{ and } y = K$$

117 Since we are dealing with undetermined coefficients this may be written

$$Ax^2 + By^2 + Dx + Ey + 1 = 0$$

thus eliminating one of the constants. This equation can be solved by the Method of Least Squares, but a solution involving four unknown quantities is so laborious that it seemed desirable to eliminate some of them if possible. By inspection of the preliminary curve it appeared that the center of the ellipse would fall on or near the point $P \equiv (30,\ 0.50)$. The coördinates of the center of the ellipse, expressed in terms of the unknown constants, are $x = -\frac{D}{2A}$ and $y = -\frac{E}{2B}$.

Hence

$$-\frac{D}{2A} = 30 \text{ and } -\frac{E}{2B} = 0.50$$

$$D = -60A \qquad E = -B$$

Substituting

$$Ax^2 + By^2 - 60Ax - By + 1 = 0$$

118 Inserting values of x and y from Table 1 we have fifteen observation equations from which by the Method of Least Squares are obtained the two normal equations whose solution gives us the values of A and B for the empirical equation sought. After simplifying we have

$$x^2 - 60x - 7023.78\,(y - y^2) = -1872 \qquad [4]$$

which is the most accurate expression of the assumed form obtainable from the fifteen examples upon which it is based. The curve of this

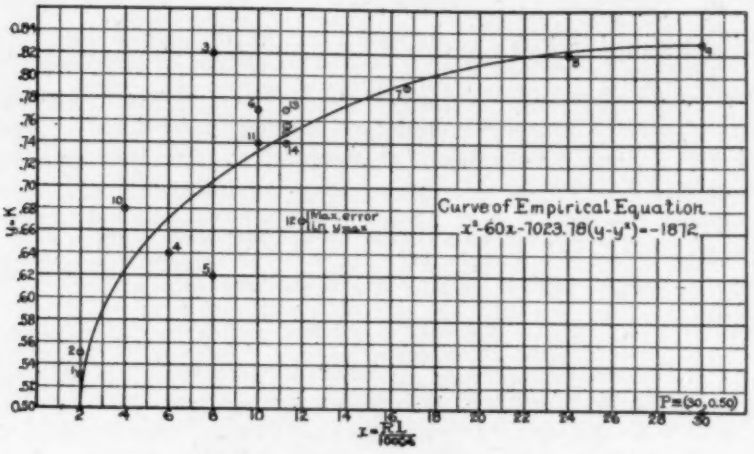


FIG. 14 CURVE FOR DETERMINING K IN EQUATION 3

equation is plotted in Fig. 14 from $x = 2$ to $x = 30$, or from $\frac{RL}{c} = 2000$ to $\frac{RL}{c} = 30\,000$. This is the full scope of the equation and it must not be applied to values of $\frac{RL}{c}$ outside these limits. It is believed that nearly all practical cases will lie well within them.

119 The observation points are also shown in Fig. 14, and it should be noted that in individual cases the true value of K may vary considerably from that shown by the curve without affecting y_{max} very greatly. For instance point No. 3 lies farther from the curve than any other and yet the error in Example 3 as shown in Table 2 is only 2.4 per cent.

120 Table 2 gives a comparison of the values of y_{max} deduced by each of the three methods thus far developed.

121 My values were determined by substituting in Equation [3] the values of V'_1 and V'_{max} from Table 1 and K from the curve of Fig. 14. The value of V'' thus obtained was substituted in Equation [7'] which was then solved for y_{max}. By the "author's method" in the table is meant the method which he uses in Par. 85 to 88 where he substitutes V_{max} for V_2 in Equation [7]

122 In using Mr. Harza's Equation [20] I have given V_2 the value of the final quiescent velocity, which I understand to be the correct interpretation.

123 Table 2 shows that my method gives an average error of 1.5 per cent, Mr. Harza's method 5.1 per cent, and Mr. Johnson's 12.2 per cent. My maximum error is 8.4 per cent, Mr. Harza's 27.3 per cent, and Mr. Johnson's 33 per cent. That this method will in

TABLE 2 COMPARATIVE RESULTS BY VARIOUS METHODS PRESENTED

Ex.	True value of y_{max} by Arith. Integ.	Writer's Method using equation [7] with $V_2 = V^*$			Author's Method using equation [7] with $V_2 = V'_{max}$		Harza's Method using equation [20] with V_2 = final quiescent velocity			V_2 for Equation [20]
		y_{max}	Error per cent +	Error per cent −	y_{max}	Error Per cent +	y_{max}	Error Per cent +	Error Per cent −	
1	9.56	9.44	.....	1.2	11.28	18.0	9.88	3.4	.....	14.18
2	14.84	14.50	.....	2.2	17.90	20.5	12.08	.....	18.7	14.80
3	10.78	10.52	.....	2.4	11.15	3.4	10.61	.....	1.6	14.57
4	11.62	11.71	0.8	.....	12.51	7.7	10.13	.....	4.2	15.20
5	12.61	12.85	1.9	.....	13.75	8.9	11.72	.....	7.1	15.65
6	15.64	15.46	.....	1.1	16.87	7.8	15.64	0.0	0.0	14.68
7	17.69	17.76	0.4	.....	18.92	7.2	17.54	....	0.8	15.06
8	15.10	15.12	0.1	.....	15.94	5.6	15.23	0.9	.....	13.92
9	17.60	17.60	0.0	0.0	18.50	5.0	17.63	0.2	.....	14.42
10	9.00	8.73	.....	2.9	10.56	17.2	9.14	1.6	.....	14.89
11	14.74	14.71	.....	0.2	15.67	6.3	14.17	.....	3.9	13.75
12	22.95	24.92	8.4	.....	30.55	33.0	16.69	.....	27.3	15.21
13	8.80	8.72	.....	0.9	9.90	12.5	8.48	.....	3.6	14.00
14	22.77	22.77	0.0	0.0	26.20	15.2	22.12	.....	2.9	15.00
15	18.07	17.99	.....	0.4	20.70	14.5	18.00	.....	0.4	15.00
		Average error = 1.5% Max. " = 8.4%			Average error = 12.2 % Max. error = 33.0 %		Average error = 5.1% Max. " = 27.3%			

general give much more accurate results than either of the others is, I believe, obvious. It is, however, more complicated and its use involves some difficulties which can best be explained in the solution of a practical example.

124 Suppose, for example, we wish to design an open surge-tank for a plant which is to develop 40 000 h.p. at full load under a head of 372 ft. The penstock is to be 8000 ft. long and under a 20 per cent sudden increment of load the surge must not exceed 25 ft.

125 The full load velocity in the penstock must be determined from considerations of economy in installation and operation which need not be discussed here. Let it suffice to say that in this case we have determined upon a full load velocity of 11.5 ft.per sec. At this velocity it will require a penstock approximately 12 ft. in diameter to supply the full load demand under the working-head. According to Williams and Hazen's formula for riveted steel pipe ten years old the friction loss will be 24 ft. for a velocity of 11.5 ft. from which $c = 0.1815$.

126 In order to cover the worst conditions we must design for a velocity change from V_1 to V_2 with the highest actual value of V_2. Consequently V_2 must be the full load velocity or $V_2 = 11.5$. We must now determine V_1 such that when the load is increased 20 per cent the steady velocity in the penstock after all surging has ceased will be 11.5 ft. per sec. Tabulating values

$V_2 = 11.5$ ft. $\quad L = 8000$ ft. $\quad y_{max} = 25$ ft.
$c = 0.1815 \quad h = 372$ ft. $\quad$ Load change $= 20$ per cent

Now

$$V'_1 (h - cV_1^2) = V_2 (h - cV_2^2)$$

and

$$V'_1 = 1.2\ V_1$$

Substituting

$$1.2V_1 (372 - 0.1815\ V_1^2) = 11.5 \quad (372 - 0.1815 \times 11.5^2)$$
$$V_1 = 9.37$$

The problem now is to find R.

127 In order to apply Equation [7'] we must first find V'' which, however, according to Equation [3] depends upon K which in turn depends upon R. Obviously in order to make a correct solution we must assume the correct value of R in determining V''. It is practically impossible to do this except by a succession of trial values and hence it will usually take several solutions to determine the true value of y_{max}. In fact I regard this equation as of more value in testing values found by other methods than for original work.

128 In order to get as close a trial value of R as possible let us take Mr. Harza's Equation [20] and solve for R. Substituting

$$625 - 744 \times 25 = -2R \left\{ \frac{8000}{64.4} \times 44.45 + \frac{372 \times 49.49}{3.14\sqrt{R}} \times 2.13 \right\}$$
$$R = .33$$

Using this as a trial value

$$\frac{RL}{1000\,c} = \frac{.33 \times 8000}{1000 \times 0.1815} = 14.5$$

From the curve Fig. 14, $K = 0.778$.

We must now determine V'_{max}. It may be figured from either V_1 or V_2. The equations are

$$V'_{max}\ \frac{1.2\,V_1\,(h - c\,V_1^2)}{h - c\,V_1^2 - y_{max}} \text{ or } \frac{V_2\,(h - c\,V_2^2)}{h - c\,V_1^2 - y_{max}}$$

In either case $V'_{max} = 12.085$. Then from Equation [3]

$$V'' = 11.244 + 0.778\ (12.085 - 11.244) = 11.898$$

From Equation [7′]

$$y^2_{max} = \frac{0.33 \times 8000}{32.2} \times 6.3908 + 95.225$$

$$= 523.97 + 95.225$$

$$= 619.20$$

129 The correct value of y^2_{max} according to our hypothesis is 625. Therefore the right-hand side of the equation is 5.8 low showing that the value of R should be increased a little. Since the increase will not affect K appreciably we must arrange to add to the first term only. This term varies directly as R and hence the new value of R should be

$$R = \frac{0.33\ (523.97 + 5.8)}{523.97} = 0.3337$$

Solving again

$$\frac{RL}{c} = 14.65$$

from which $K = .779$ and $V'' = 11.90$

$$y^2_{max} = \frac{0.3337 \times 8000}{32.2} \times 6.3908 + 95.23$$

$$= 529.77 + 95.23$$

$$= 625$$

$$y_{max} = 25$$

which is the desired value.

130 Integrating the curve arithmetically with $R = 0.3337$, I find $y_{max} = 25.08$ showing that the value found by Equation [7′] was about 0.3 per cent low. R should have been a trifle less and it may

be that the value $R = 0.33$ found by Equation [20] is nearer the truth than my own. Certain it is that both equations are very close and an error of 0.3 per cent is not worth considering. From a practical standpoint the re-adjustment in the value of R was scarcely necessary. It was made to illustrate the best method to pursue in cases where it is necessary to make a correction.

131 In view of the close results shown by Mr. Harza's Equation [20] in the problem just solved and in the other fifteen examples I think it but justice due him to say a word in praise of this equation. For an equation containing at least one, if not more, arbitrary assumptions it shows a surprising degree of accuracy under a wide variety of conditions. In every instance it is more accurate than the author's Equation [7]. The average error is less than half that of Equation [7]. In fact were it not for the two instances where the error runs up high I should consider my own method an unnecessary refinement. There are two cases, however, where the error is 19 and 27 per cent respectively, showing a possible deviation which ought to be provided against. The former case is a perfectly normal one although in fairness it should be said that the latter case is not a normal one and would probably never occur in practice. Its introduction, however, is no more unfair to Mr. Harza's method than to the other two. Another point of merit in this equation is that it is the only one so far presented which states in simple terms and with fair accuracy the relation between the maximum surge and the initial and final quiescent velocities.

132 In conclusion I wish to emphasize the fact that throughout my discussion V_2 has had but one meaning, namely, the final quiescent velocity in the penstock after all surging has ceased. This interpretation is based upon the author's statement in Par. 35 that "$V_2 - V_1$ is the ultimate change in velocity due to the load change." Hence V_2 must be the ultimate velocity due to the load change and assuredly that is the final quiescent velocity. To my mind no definition of the word "ultimate" can possibly justify its application to an intermediate momentary value during the surge such as V'_{max}.

AUTHOR'S CLOSURE

THE AUTHOR. The purpose of my little monograph on Surge Tanks was to awaken some much-needed enthusiasm on the subject, and to call forth discussion and adverse criticism, in the belief that the resulting matter when coördinated would not only add to my own

store of information, but present a collection of ideas sufficiently varied to enable any interested student, to separate the wheat from the chaff and arrive at a correct understanding of these complicated phenomena. The outcome thus far is highly gratifying, but still a long way from the result sought. Mr. Larner's work particularly is of great practical value.

2 My paper was prepared at very short notice and is decidedly incomplete when considered in the light of a text-book, and deficient in what might be instructive examples as to the limitations of its equations. After persual of the work of my critics, I want to say at the outset, however, that I find absolutely no reason for altering any of my equations, unless it were to make them universally exact, a difficult task which has not been done in the discussion, as I shall proceed to show. Equations 7 and 12 called forth the most comment, and therefore I shall devote myself to a short general discussion of these two equations in the light of criticism. They are just what they purport to be, nothing more nor less, and their accuracy according to the preliminary assumption is unquestionable. Both of them assume a constant draft velocity as the superior limit of the integration, and hence have only an indirect value when applied to an actual case where the draft velocity varies (particularly in the simple tank regulation) according to the action of the governor in its effort to maintain constant power. I described this effect at great length, and yet the language of my critics in some passages would convey the impression to any but a very careful reader, that I had overlooked this point and attributed to my equations a perfection entirely unwarranted by facts.

3 Mr. Larner says in Par. 66 that he is unable to perceive the logic of developing such an equation as 7, and yet he offers nothing better which is usable; and he adopts Mr. Harza's Equation 20, which I shall show later gives less satisfactory results for some cases than Equation 7. I have intended to claim no exact application for this equation, which avowedly neglects the action of the governor (Par. 37). Mr. Harza does claim rather exact application for his Equation 20. The fact of the matter is that this Equation 7 is extremely useful, and sufficiently accurate if judiciously used by one who makes a study of the subject and learns to know its value. Professor Church has proceeded logically to show its limitations, and this is a good step. But as will be shown, he has not worked the equation to its best advantage, because he has not mentioned the fact that the final quiescent velocity, after all surging has ceased, may be *known before*

hand as well as the immediate value of 8 ft. per second. Mr. Harza's novel way of introducing the governor action is undoubtedly a valuable step in advance of my work, and is in the right direction.

4 Equation (12) deals primarily with a *velocity change* between two limits V_1 and V_2. It is therefore directly applicable only to the particular large velocity change for which the design is made, and beyond which one is supposed to care nothing about close regulation. When the design is adequate the surge is often a very small proportion of the head, and obviously should be so, because otherwise one would have to provide a much larger water wheel than necessary for the purpose of holding up full load during the depression of the surge, and the wheel would afterward be continuously operating at a large reduction of gatage and at poor efficiency. Therefore, it often happens, especially in designs for large plants, that the superior limit of V_2 is not largely different from the value to which the velocity instantly changes when more load is demanded; it is always more accurate, however, as I pointed out in Par. 55, to increase this value by multiplying it by $\frac{h}{h-y_1}$ before introducing it in the equations.

5 Mr. Larner has made this off-hand statement of mine the bone of contention in his arguments; I want, therefore, to point out the practical reasonableness of my statement, as well as its accuracy, and thereby show that the consideration of the change of gradient, of which Mr. Larner makes such a point, does not naturally or necessarily enter into the question. Curiously, the same error of neglecting the change in gradient which he attributes to me, he himself makes, at least partially, when he states that its effect upon the surge in the simple tank is practically nil (Par. 62*a*). (See also Par. 35, 73 and 74.)

6 As stated in my Par. 55, "h" is the "working head," which means to me the net head for a velocity of V_1 although to Mr. Larner it seems to mean the gross head. I myself never use the term in that way. Suppose the full load capacity of our plant is 50 000 h.p. and one wants to provide for a maximum sudden increment of load of 5000 while running at 45 000 h.p. output. The first step is to make a computation for the working head at 45 000 h.p. output. The actual value of "h" within reasonable limits does not ordinarily affect much the value of $\frac{h}{h-y_1}$ if "y_1" is not more than 5 per cent of h, as is often the case. One may easily compute "h" by trial with more accuracy than that with which one can foretell the friction value, and with

much more refinement than is perhaps warranted by one's knowledge of the future maximum load change and the water wheel efficiency.

7 Having selected "h," then V_1 follows from it, or one might say is selected with it, for the working head is the gross head less CV_1^2. Now, if the head and wheel efficiency did not vary, the value of V_2 would be $\frac{50}{45} V_1$. It is customary to neglect the change in wheel efficiency, but it is always more accurate to increase V_2 thus found by assigning to it a larger value, which is occasioned very suddenly in the differential regulator on account of the dropping off in head an amount "y_1" which one selects. The correct value of this increased V_2 to be used in Equation 12 is, as stated, $\frac{V_2 h}{h-y}$. (See Mr. Larner's Par. 94.)

8 In all my work V_2 is not *any particular velocity* but simply the *superior limit* in my integrations under the hypotheses. The use one makes of it in an actual case depends upon individual knowledge of the question, and is purely a matter of judgment. In the premises (Par. 85) I have assigned very plainly a value of $V_{max} = 15$, as this superior limit for the numerical examples, merely as an illustration of the fact that V_2 may be given that value. It naturally gives more accurate results for Equation 12 than for 7, and I am perhaps subject to criticism for using this value in Par. 87 instead of some smaller value, but not knowing the correct figure to use for the best result except by arithmetic integration, I used that figure as an illustration, expecting the result to be on the safe side and not far from the truth. In applying Equation 12, in Par. 91, I have used the value of 15 for V_2 as plainly as I am able to state it, and V_2 in the premises is taken as V_{max}, which may always be obtained by multiplying by $\frac{h}{h-y_1}$ as stated.

9 I am dwelling upon this at length because in his able discussion Mr. Larner has misinterpreted the ideas which I set forth. His paper is a valuable one because it is accurate, so far as I have checked it, and it gives a good airing of the subject, but nearly all of his work is curiously illustrative rather of improper and inconsistent uses of the equations than of the incorrect use for the best approximate results. For example, he states in Par. 105 that if the correct value for V_{max} be used as the superior limit in Equation 12, then the resulting values will be correct; by which he means correct for his case 4, as one may easily determine by reading his development of a "true

theoretical equation" (Par. 91). If then V_{max} is determined as I pointed out in Par. 55, and used as the superior limit in Equation 12, the resulting equation is a "true theoretical one," according to Mr. Larner, and indeed I myself nearly concur in this opinion. Referring now to Par. 49, he states that Equation 12 is applicable only to Cases 1 and 2 (see Par. 80). Also, in his comparative example by arithmetical integration, illustrated in Fig. 4, he gets a value for V_{max} of nearly 16, and still he claims that because by so doing he attains a surge of 15.3, my formula is wrong, since with $V_{max} = 15$, I obtained 10 ft. as the value for y_{max}. The only conclusion I can reach from this is that the equation will not apply where it is supposed to apply, as stated by Mr. Larner in Par. 86.

10 Mr. Larner's reference, in some instances to the "author's contention," etc., refer to hurried answers to questions asked in correspondence. So far as I know my statements were consistent and if not I could possibly explain them. In Par. 95, Mr. Larner states that my Equation 12 is the same as what he terms his Equation 2, provided one uses the correct value for V_{max}. Inasmuch as there seems to be no particular reason for not using the correct value, the originality of Equation 2 seems to lie only in that it contains rather a condensed means of computing the various power outputs from the existing heads, allowing for friction; which is elementary and not an essential in Equation 12.

11 As to Mr. Larner's criticism (Par. 83) of my formula for port area, I agree that it is not theoretically exact except under the hypotheses stated in Par. 48. Here again, however, he writes out an equation which is exactly the same thing as my own, if one substitutes V_{max} for V_2, and therefore his equation contains the same fault as my own.

12 It is quite true that the area by this formula is a little large, but as stated in Par. 54 it is better to err on that side. In order to produce an ideal set of curves for the problem in Par. 91 the port area would have to be reduced by perhaps one-twentieth. The accompanying curves show the result of a design as in Par. 91 with the port area thus reduced.

13 For this work the area of the stand pipe was taken as 25 sq. ft., and $R = 0.25$ to facilitate the computations; also the port area was made sufficient to pass 60 c.f.p.s. under a head of 1 ft. The choking head at the entrance to the conduit is neglected, as is quite reasonable in this case if the surge pipe is slightly flared below the ports where it joins the conduit. Inasmuch as Mr. Larner has not produced a set

of curves illustrative of the problem he was studying, I thought this addition to his work would be interesting. His curves all represent possible cases and are therefore very instructive—but they have no value as a check on Equation 12, because his premises are different from those assumed for the problem as computed with the formula.

14 An examination of the surge pipe pressure curve shows quite a wide departure from the theoretical rectangular curve assumed in Equation 12. One might, therefore, expect considerable error in its application to such a curve. This is not the case, however, for

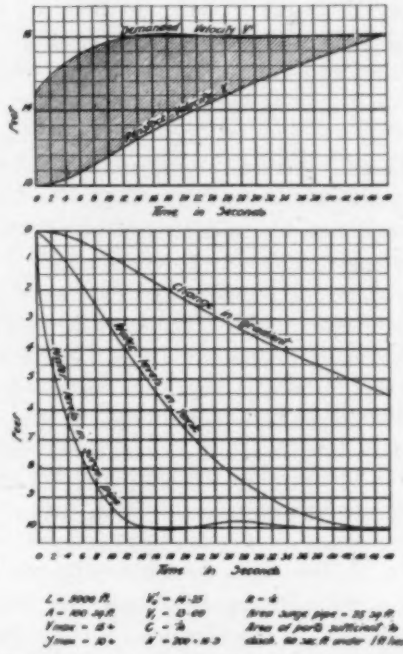


FIG. 1 CURVES ILLUSTRATIVE OF PROBLEM REFERRED TO IN PAR. 12

usually no appreciable error is to be discovered in a practical working example. The reason for this apparent contradiction seems to be very well explained by the fact that the time is longer than that found for the hypothetical cases, and hence the lagging in the first part of the cycle is recovered by an extension of time at the end. Furthermore, in the actual case there is a relief from some of the work contemplated in the hypothetical case due to the fact that the demand is not instantly changed to its maximum value. The shape of the pres-

sure curve here illustrated is strictly typical of an ideal curve, as I have found in dozens of cases. There is always noticeable that arch effect in the horizontal portion of the curve.

15 I want to say in justice to Mr. Larner, lest the inference be drawn that I undervalue his excellent work, that in all instances his conclusions and adverse criticism are justifiable from his point of view, and in many instances from my own, on account of a vagueness and incompleteness of explanation on my part, which I can clearly see in the light of his analysis. He has tried to set forth not so much his own opinion of what I have intended to show, as his idea of how I should have expressed myself to make my meaning clear to one less qualified to discern than himself. At least, that is my idea. It is none the less necessary to point out the intent of my meaning, in a forcible way, both, in justice to myself and for the further clearing up of a number of quasi-disputed points.

16 It is gratifying to note the close agreement between Mr. Harza's formula and the preliminary ones in my paper, and it is to be hoped that his work of analyzing and checking will be continued to include the differential principle, and the use of compressed air.

17 I have not given the attention to the simple stand pipe that Mr. Harza has, because I have always regarded the methematics relating to it as simply a step toward the discovery of a regulator which would meet more perfectly the requirements of practice. Such a regulator, I believe, is to be found through a study of the "differential principle," and I have yet to be convinced that the regulator I propose is not far better than the simple stand pipe, despite Mr. Harza's argument somewhat to the contrary. His reasoning, in the abstract, I believe is sound, but it shows a possibility of modification after more detailed study of the question with the essential aid of definite dimensions, which may be found by tedious arithmetic processes of integration. The *tendency* to produce worse regulation conditions by the introduction of a resistance between surge tank and conduit certainly exists, and is due to conditions very much as stated by Mr. Harza—an appreciable bad effect upon regulation never gets to the point of appearing, however, because lost sight of when compared with the manifest advantages. Mr. Harza shows a realization of the possibility of this condition in Par. 47. I have no doubt that further study of the question on his part will largely clear up this slight difference of opinion.

18 I have never hit upon a method of approximation, such as Mr. Harza's Equation 20, which I regarded as accurate enough to

publish, and was compelled to leave the effect of the governor's action to the judgment of the designer in selecting his value of $V_2 - V_1$. I regard his equation of especial value as proving how little practical effect the governor's action has, in some instances, *when the other conditions* for good regulation are satisfied.

19 Another reason for my not giving much attention to the governor action is that it does not enter into the problem in the same way when the differential principle is used. In this case, it is necessary only to select a value for V_2 in excess of that which one *expects* to represent the maximum load change in the future plant, as pointed out in Par. 55.

20 As to the argument (Par. 36) against catching all the energized water, I should like to point out two things: First, it does not appear clear to me that curtailing the upward surge, whose maximum occurs for rejected loads, puts any limit on the downward surge, whose maximum occurs for demanded loads, except where a super-added or a perpetual wave exists. The latter condition is possible where the differential action is omitted, and where one is really designing for minimum size of tank consistent with regulation requirements. In that case, water would continue to spill at regularly recurring intervals all day, or until another load change happened to occur in such a way as to fit into the existing wave and smother it. Mr. Harza had in mind the "super-added" wave, but even this is far more likely to occur when the differential action is omitted. It was probably not intended to imply the possibility of such an obviously poor design when arguing its advantage.

21 Second, the saving of water at the rare intervals of "full load rejected" is not of much consequence, to be sure; but the saving of the cost of construction work necessary to catch and lead away a possible large discharge due to a shut-down is of such importance as to weigh decidedly in the balance against the additional cost of surge tank to catch and hold all the energized water. Allowing the water to spill over has a damping effect upon surge waves for rejected loads, as he states, but this can be accomplished much more effectually by means of the differential principle without waste of water and usually at a saving in cost besides. It is possible, through such a regulator, if designed especially for maximum damping effect, to make the surge wave strictly and absolutely dead beat; and the importance of a thorough study of this principle cannot be overestimated.

22 Referring now to Mr. Harza's Par. 49, I want to say in justice to Mr. Warren's valuable paper that my remarks were not intended to

attain the dignity of a criticism, but merely as introductory to a line of argument which seemed to be new to him. I am still of the opinion that a properly designed regulator, or system of regulators, is actually "a complete remedy for the troubles in speed regulation caused by the excessive inertia of a water column." Mr. Harza means, I judge, that the surge tank cannot be made a complete remedy for variation in head—but it is obvious that a moderate amount of head variation does not at all mean "troubles in speed regulation;" for otherwise, where do his own equations lead? His criticism of my work is very creditable and fair, it seems to me. The values obtained by my Equation 7 are not as close to the truth, in the instances cited, as those obtained by his Equation 20, as might be expected; although, as I shall show later, Equation 7 gives closer results when applied to Professor Church's examples.

23 These problems contain an inordinate amount of friction, which is rarely more than 10 per cent for a correctly designed plant, and often as low as 5 per cent, but I use them for the sake of illustrating the inconsistency of Mr. Larner in making the statement that friction may be disregarded (Par. 52) and at the same time calling me to account (Par. 89–95) for disregarding the change in gradient, where this mistake was due to his misinterpretation. In Par. 89 he intimates that persual of my paper would disclose the fact that the superior limit of the velocity in my equations, which I have called V_2, is invariably intended to be applied to the constant quiescent velocity, or the final value of the velocity after all surging has ceased, and steady running conditions are once more established at the new load value. Granting for the moment the reasonableness of such a conclusion, one notices that Professor Church must have used something more than "ordinary care," for he does not substitute such a value for V_2. If he had done so, however, the results would have been fairly good in the examples cited, and therefore in these instances it would have been better to do so, but not always by any means. V_2 is naturally the "ultimate" velocity under the assumption of a constant draft velocity (see Par. 35).

24 Let us adopt Professor Church's nomenclature for this little study, with the addition of V_3 which he does not use but which I need as a symbol for the steady velocity under the new load after all vibrations have ceased. Let H be the whole head; then from the principle of constant power we have, neglecting the change in wheel efficiency

$$(H - CV_1^2)\, V_0^1 = V_3 H - CV_3^2 \qquad (23)$$

also $C\ (V_1^2 - V_3^2)$ = the net change in gradient or the difference in head before and after the surge, which one may call Δh.

25 Now a very little thought at this juncture will disclose the fact that y_{max} must be at least as great, in all cases, as Δh; it is also nearly as plain, I think, that y_{max} is always greater than Δh; however, for my present purpose, I am content to stop with the former statement which admits of no argument, for I propose to solve for V_3 in each of Professor Church's three cases, and to show that Δh is

TABLE 1

	V_3	Δh	V_2	y_{max}
Case 2	3.88	84.95	4.35	81.50
Case 3	6.10	25.00	6.23	24.60
Case 4	7.60	10.60	7.56	11.60

greater than his y_{max} in all except the last case. This, of course, leads to but one conclusion, which is that his values are too small. I do not doubt, however, that they are as accurate as one could figure them by the methods adopted by him, and they are undoubtedly near the truth, but not quite so near in two cases as could have been obtained directly by this equation (23), just written. Of course, this equation is of no value whatever in determining y_{max} except where $\frac{c}{R}$ is so very large that y_{max} is not sensibly larger than Δh.

26 By trial in Equation 23, the following values are found for V_3 and Δh for the three cases and I have also listed for comparison Professor Church's values for his V_2 and y_{max}.

Example (case 2):

$$H - CV_1^2 = 80$$
$$V'_0 = 8$$
$$H = 180$$
$$C = 1.00$$
$$640 = 180\ V_3 - V_3^3, \text{ from which by trial } V_3 = 3.88$$
$$\Delta h = (100 - 15.05) = 84.95$$

27 Let us now list the values for y_{max} as obtained from my Equation 7, putting V_3 (*known before hand*) as the superior limit, and compare them with the values for y_{max} obtained from Mr. Harza's Equation 20, putting his $V_2 = V_3$ above. He states in Par. 23 that the upward surge is always less than the value given by his Equation

20. Now these are examples of upward surge, and yet it is seen that Equation 20 gives not only smaller values than the true y_{max} but even smaller than Δh, which is itself necessarily at least a little less than the true y_{max} in all cases. It is also seen that the much buffeted Equation 7 gives closer results in these instances than the more exact equation of Mr. Harza's.

TABLE 2

	Equation 20 $y_{max.}$	Equation 7 $y_{max.}$	Δh	Professor Church y_{max}
Case 2	46.60	91.4	85.0	81.5
Case 3	16.00	27.9	25.0	24.6
Case 4	8.75	13.0	10.6	11.6

28 The correct values for y_{max} probably lie between Columns 2 and 3. These computations are all made on an 8 in. slide rule, but are presumably otherwise accurate. The comparison is not made for the purpose of discrediting Mr. Harza's Equation 20, which is a valuable one when judiciously used, just as my own are. I doubt if Mr. Harza himself would make the mistake of using his equation for such unusual cases as 2 and 3, and yet he authorizes its use, to be strictly literal, in Par. 28. I am therefore justified in thus drawing odious comparisons, particularly in view of Mr. Larner's statement (Par. 52) that the consideration of friction sometimes reduces the figured surge. As a matter of fact, it seems to me that friction must always increase the surge (as stated in Par. 23), though oftentimes a negligible amount where $\frac{c}{R}$ is relatively small, in which case Mr. Harza's Equation 20 would probably give excellent results. There is and perhaps can be no hard and fast set of formulae which will accurately cover all cases. One must study the question from all points of view, and work out many examples by arithmetic integration before attempting to use one's judgment. The formulae are all useful to one used to their limitations, and in any event, when all is said and done, one should always check up by means of arithmetic integration before proceeding to build a regulator.

29 In Par. 48, Mr. Larner states that I have "*contended*" that the governor effect is not very important. As I re-read my Par. 24-28 inclusive, and also Par. 56, I wonder if others will draw the same

inference. It is quite conceivable that one might think from the language of Par. 37 and 55, that this important matter had been slighted; I plead guilty to carelessness of expression, but not to intent to deceive. Proceeding exactly as I did in the numerical examples, putting $V_2 = V_{max}$ one incidentally eliminates all governor effect on the surge, and in the case of the timple tank one has some to spare besides; also I wanted to bring out forcibly the point that oftentimes no disastrous results are obtained by substituting what Professor Church terms V'_0 or V_3 for V_2, as illustrated by his case 4 and carefully explained by Mr. Larner himself in Par. 75 and 76.

30 One always knows beforehand the values of V_{max} and V_3 (since y_{max} is selected to be the result of the design). By substituting each in turn if necessary for V_2, one can usually, in a *practical case*, fix the true value of the surge within limits, and thus be sure of being at least safer than by the use of any formula that purports to include all the variable quantities and only partially does so. It should not usually be found necessary to make any material alteration in tank dimensions after checking up by arithmetic integration, as I have repeatedly proved. Therefore, for my own use, I prefer a simple equation like 7, which has the advantage of being fairly accurate for what it purports to show, to a more exact one which only partially does what it claims, always leaving one in some doubt as to the limits between which the true result lies. The following figures will assist in making a little more careful study of the numerical example of Par. 87, in order to help demonstrate the usefulness of Equation 7, when applied with discretion.

31 First, let us solve for V'_0 and V_3; we have

$$V_1 = 13$$
$$V_{max} = 15$$
$$h = \text{working head for } V_1 \text{ or } V'_0$$
$$H = h + C\,V_1^2 = 216.9$$
$$h \text{ for } V_{max} = 190$$
$$190 \times 15 = 216.9\,V_3 - \tfrac{1}{10}\,V_3^3$$
$$V_3 = 14.55$$
$$190 \times 15 = V'_0 \times 200$$
$$V'_0 = 14.25$$
$$\Delta h = \tfrac{1}{10}\,\overline{(14.55}^3 - \overline{13)}^3 = 4.25 < y_{max}$$

Equation 7, with $V_2 = V_3$, gives y_{max} = about 7.75
Mr. Harza's 20, with $V_2 = V_3$, gives y_{max} = about 8.25
Equation 7, with $V_2 = V_{max}$, gives y_{max} ± about 10.00

32 None of the values are right, but Mr. Harza's value is without doubt the nearest, though presumably small. If the correct value is about 9, then it is apparent that a tank somewhat smaller than 34 ft. could be used, and therefore my numerical comparison between the simple tank and the differential is rather unfavorable to the former; just how much I cannot say without more work than is possible within the time at present available. Pressure of time made it necessary in the first instance to close my paper almost before I had started any numerical illustrations, and thinking these power computations for different heads elementary, I contented myself with a hurried illustration, believing that any one who went into the subject sufficiently to understand it would have no difficulty in getting some good out of my formulae; for the others, probably no amount of explanation would have sufficed, and the explanation might have been fully as misleading to them as the work I did.

33 The value of R is very sensitive to the magnitude of V_2 and an absolute computation for it is perhaps impossible by any reasonable process; but y_{max} is not so sensitive, and y_{max} is really what one wants to foretell within reasonable limits. It must be remembered also that V_2 is only a guess in the first place, supposedly of sufficient size as compared with V_1 to represent a maximum load change. My study of the vagaries of Equation 7 have not been extensive enough to warrant my writing at any great length. My belief in the superiority of the differential scheme has led me to work with equation 12 for the most part, much to the neglect of Equation 7. I believe, however, that it is a valuable equation for use in connection with Mr. Harza's Equation 20, and probably a combination of the two can be worked out which would be quite accurate for all practical cases.

34 Before closing, I want to say a word about Mr. Larner's statement that for abnormally large loads the simple tank is more "elastic" (see Par. 56, also 62*c*); there is not much in that, practically, because one designs for a certain depth of water at part load in either case, the depth being sufficient to pick up full load. If the load change exceeds that for which one designs, as a maximum, the simple tank will become empty and cease to regulate, just as will the differential form—and at any rate one should fix this maximum load as large as that for which one cares to regulate. Considering two tanks of the same size, the maximum load change under which the simple tank would become empty would be much smaller than that which would empty the differential regulator, and therefore the latter would

be much more "flexible" and "elastic" for all load changes between these two maxima.

35 Another important point in favor of the differential form which I inadvertently omitted to mention, is that since the maximum surge is less a smaller water wheel is required to pick up full load as the head recedes; this works two ways to advantage: the units are less expensive in the first place, and the wheels operate continuously at larger gateage and hence at better normal or average efficiency.

36 There is one important point affecting the accuracy of arithmetic integration when applied to these surge waves, which I have only just touched in Par. 89, and that is the choking head at the entrance to the surge pipe or tank This connection leading out of the conduit to the regulator is necessarily of small area as compared to the tank, and losses of head occur which are neglected in all the formulae and are often overlooked in the arithmetic work. This loss, together with the friction in the riser, in the differential scheme, is not of great importance for small velocity changes, but for a shut-down or a large change of load it may become one of the most important factors. On this account it is evident that Mr. Larner's illustrated examples cannot be strictly correct, although sufficiently so for the purpose intended. I have found it sometimes unsafe to neglect these losses, but like many other things one has to bear them in mind when applying the equations. From the foregoing argument it is apparent that the simple tank is usually not altogether free from differential action, and hence the pressure curve is a double line for large changes of load.

37 Lack of time forbids any useful comments on Mr. Knowles' problems, which present questions altogether too complicated for intelligent discussion in anything like a casual manner; perhaps the method of trial and error by actual experiment is the only way to determine correct solution, and Mr. Knowles is following that line.

38 It has occurred to me that some confusion may result from the foregoing careless use of the expression V'_{max}. This has been intended to refer to the maximum velocity drawn by the wheels, and would be perhaps more properly designated as V'_{max}, The real V_{max}, or the maximum velocity, in the conduit for demanded loads, is always greater than V'_{max}, because acceleration of the long column continues some time after y has reached its maximum, and after V' has begun to decrease. When V_{max} exceeds V_3 by an amount greater than the difference between V and V_1, there is grave danger of a per-

petual wave. For the same values of y_{max} and $V_3 - V_1$, the excess of V_{max} over V'_{max} is less in the differential regulator than in the simple tank. The time when V reaches its maximum value compared with y_{max} is clearly shown in the example of an augmenting wave.

39 In Par. 91 I used the value of V'_{max} as the value for computing the capacity of the tank for a shut-down. Inasmuch as the actual conduit velocity will exceed this value, and since a shut-down may occur at any time, and the formula is not sure to give an outside figure for y_{max}, it is not thought an excessive precaution to use the value of V'_{max}. Mr. Larner did not see any sense in this, however (see his Par. 88), possibly on account of insufficient explanation on my part. I relied on the text to clear up all these points, and attempted to describe the cycle of events with clearness sufficient to preclude any misinterpretation of the mathematics. In the criticisms no mention has been made of the perpetual wave. This, it seems to me, is an important feature to consider. There is nothing in the formulae for the simple tank computations to warn one when approaching the danger limit. I should say, off hand, that when $V'_{max} - V_{\frac{1}{3}} = V_3 - V_1$, there is every likelihood that succeeding waves will be greater than the first one, which would mean a never ceasing change of head and an extremely undesirable condition.

40 For values of $V'_{max} - V_3$, even considerably less than $V_3 - V_1$ the condition of too long life of the wave, even though the wave died out eventually, might well be brought about, making it quite probable that a succeeding new load-change would cause the waves to accumulate disastrously. When $V'_{max} - V_3 = V_3 - V_1$, it can be easily shown that

$$y_{max} = h - \frac{V_0' h}{2V_3 - V_1} \qquad [24]$$

where h is the working head for V_1.

41 Without sufficient study to warrant a decisive statement, I would at least suggest that the value of R be always made less than half that which would give this value of y_{max}. If one has the patience one can try out a number of cases by arithmetic integration, where $y_{max}h$ as values some less than that given above and see what eventually becomes of the wave. This ought to be done. It looks as though this expression would be at least a fairly correct index as to the danger limit, and it is easily applied, because V'_0, V_1 and V_3 may always be easily found as previously pointed out.

42 Until something better is developed, this value of R may

be taken as indicating the region of the danger limit. It is at least a better expression of the facts than the empirical rule which I previously offered in Par. 57. V'_{max} is, to be sure, not as large as V_{max}, but on the other hand it is probably always too large for the correct V_2 in Equation 7. It is possible that these two errors will tend to offset each other sufficiently to indicate with some accuracy the critical value of R in a practical working case.

43 Mr. Larner also at times uses the expression V_{max} for the maximum draft velocity. The penstock velocity does not reach its maximum in the first quarter cycle, and hence the only V_{max} which really exists in his theoretical studies is V'_{max}, and there should be no confusion to a thinking man. The equations are not concerned with any velocities occurring later in the cycle than the end of the first quarter.

44 Now the V_2 which Mr. Larner has introduced into his formulae not only does not occur in the first quarter, but theoretically it never occurs in a finite time. It is the ultimate value of V' after all vibrations have ceased. It is a strain upon the imagination to involve this quantity in the computations for the first quarter cycle. It can be dragged in without error, to be sure, just as Mr. Larner has ingeniously shown, but that it is misleading and unnecessary is well brought out by the fact that in working out the curves herewith submitted I never used this value at all.

45 I had no occasion even to know what it was, for the very sufficient reason that it does not exist during the first quarter cycle which I was studying. A casual reference to Par. 6 will indicate the normal and rational way to begin a surge study. V_1 has to be known. V'_0, a function of the load change, is the same as my V_2 when the latter is assumed constant for the sake of integration. V_3, or Mr. Larner's V_2, never even appears as such during the whole arithmetic integration—and it is rather plain therefore that all of his discussion about the effect of the friction upon V', etc., etc., which results directly from appropriating this V_3 and mixing it up needlessly in his equations might as well have been omitted.

46 The introduction of V_3 involves nothing whatever but an elementary power computation, and it has no proper place in the theoretical work. Any one who chooses may compute it easily, for any rational purpose. I often do so for the sake of substituting it for V_2 in Equation 7, not because it has any real theoretical place there, but because I have found by experience that this value represents a convenient working mean between V'_{max} and V'_0, which usually gives fair results when substituted for the upper limit in Equation 7.

47 The value for y_{max} thus found is usually low, but close in a practical working case, as Mr. Harza's comparisons show. As before pointed out, one may also get a value of y_{max} greater than the true value, by putting $V_2 = V'_{max}$.

48 It is interesting to note, in view of the importance assigned to Mr. Larner's V_2, that in all his elaborate arithmetical studies for Case 4, represented graphically, this value does not even appear; and in fact, cannot appear, theoretically, in any finite time. It would, therefore be rather a laborious operation for him to prove arithmetically that he needed this value in order to proceed with his work.

49 Mr. Larner leads one to infer, in Par. 88, that Mr. Harza intends his V_2 to mean V_3 invariably. This cannot be the case, however, as a glance at his Equations 13 and 14 would indicate. In Equation 13 the value of the first V_2 was intended to be V'_0, or possibly V_3, by a stretch of the imagination. Either value would apply. It seems clear, however, that the value intended in Equation 14 for V_2 is V'_{max} because, in the time $\frac{T'}{2}$ of a quarter cycle, the velocity in the penstock varies from V_1 to V'_{max}. His work is nevertheless very instructive to me, and I realize that without the admixture of considerable judgment none of these equations can be used. Mr. Larner looks to hand-book simplicity as his ideal, but I fear that this desirable condition will never be reached. My chief criticism of Mr. Harza's work is its claim to greater accuracy than the facts warrant.

50 In the light of Mr. Larner's criticism of my work, Mr. Harza should be able to do as I have now attempted, namely: to show his real intentions where insufficiency of explanation leaves many things in doubt to any but a very careful student. Mr. Larner's attention to detail, and his insistence upon the accurate use of symbols, has opened up the whole subject to the light, and made it compulsory for me, in justification of myself, to clear up many points.

51 I think it now behooves Mr. Harza to do likewise as soon as possible. To my mind, his Equation 13 should strictly read,

$$E_w = A V_0' \frac{T'}{2} w (H - C V_1^2)$$

Equation 14 should read

$$E_a = \frac{w}{2g} A L (V'^2_{max} - V_1^2)$$

and Equation 10 cannot be expressed exactly, because it assumes a

constant V_2, just as I have done, and whether it is better to assign a value of V_3 or V'_{max} is hard to tell without much study.

52 It might be argued that the presence of V_3 in the formula is essential in order to predetermine the capacity of the water wheels after the surge. There would seem to be nothing in this, however, for the greatest tax upon the wheel is when it has to hold up full load under a head of $h - y_{max}$ in which V_3 again does not appear.

53 Several errors have crept into my paper in spite of considerable care, and I have attempted to have them all weeded out in the final printing. An important one occurs in Par. 85, where I inadvertently used the phrase "change in hydraulic gradient" instead of "change in water level," or, strictly, "change in pressure at the turbines."

54 A word in reference to Mr. Larner's laborious work of arithmetic integration will conclude my remarks. Not having the time at my disposal to check over this work, I must be content with a statement of some general conclusions. In Example 1 I note an exact agreement between the tabulated values of V_2 and V'_{max}. Inasmuch as this is theoretically impossible, and Mr. Larner has always before taken V_2 as his starting-point, one is led to question the accuracy of this example.

55 In Example 2, V_2 has a larger value than V'_{max}. This is *a fortiori* an impossible condition, because the change in tank level due to the change in gradient alone is larger than the value given for y_{max}. I have previously called this Δh. If V_2 is 14.80, then Δh must be about 15 ft., whereas y_{max} is given as 14.84. It is evident that the true value of y_{max} must be greater than Δh or greater than 15. Even if it were only just equal to 15, it would play havoc with Mr. Larner's percentages of excellence which are, as will be shown, a little misleading. A correction, for example, along the lines just indicated, would probably reverse the burden of error from the result obtained by Equation 7 to that derived from the use of Mr. Harza's equation. It appears, however, that for the most part, the tabulated results are sufficiently accurate and they show very clearly the obvious fact that Equation 7 with $V_2 = V'_{max}$ is an excellent one for practical use, because it *always gives values on the safe side*, and so near the truth that any further so-called refinement by uncertain processes is not essential. The values given by Mr. Harza's equation, although much closer on the average than my own, are not any more useful because they are too often *unsafe*, and one never knows just how much.

56 Omitting Examples 1 and 2, and also No. 12, which is practi-

cally an impossible case, I offer a table (Table 3) made up from Mr. Larner's figures, as illustrating about the degree of error in the use of Equation 7. The assumed load change is only a refined guess at best. The comparative values here given indicate the percentage of load change for which one actually provides by Equation 7, as compared with the percentage of load change intended to apply. In other words, a surge tank as designed by Equation 7 will take care of a load change, according to column 2, if designed for the corresponding load change indicated, for the various examples, in Column 1. I am indebted to Mr. Larner for his exhaustive work whereby I am enabled thus concisely to express the real error.

TABLE 3

Example No.	Per cent Load Change as estimated.	Approximate Per cent Load Change Provided for by Equation 7.	Error.
3	20	20.7	.7
4	15	16.1	1.1
5	10	10.9	.9
6	15	16.2	1.2
7	10	10.7	.7
8	15	15.8	.8
9	10	10.5	.5
10	10	11.7	1.7
11	13	13.8	.8
13	6	6.8	.8
14	15.5	17.9	2.4
15	12	13.8	1.8
Averages.....	12.6	13.7	1.1

57 The figures are exact only in so far as Mr. Larner's percentages of error in y_{max} are correct and they are made up on the further assumption that the proportionate increase in load change is the same as the proportionate error in y_{max}, which is only a rough approximation within narrow limits. Column 3 shows the practical error encountered in the use of Equation 7, expressed in precentage of part load, and may readily be compared with the total assumed change in each case. The futility of further refinement of these values is clearly brought out by the fact that it is imperative when one uses the simple tank, to work out by arithmetical integration at least *one and one-quarter* cycles in order to be assured that the pressure wave will not *live too long.* I have found this condition to exist where it could not have been foreseen by any process of reasoning available to me.

58 Mr. Larner's method of using Equation 7 will perhaps always give closer results than the method I have adopted as a *safe course*; it naturally will, at any rate, for the set of cases which he has carefully worked out. One should, however, always bear in mind, that if one adopts anything less than unity for the value of Mr. Larner's K, there is at least a good chance that the resulting design will provide for less load change than that intended. The same thing is true in his use of Mr. Harza's equation, only rather more emphatically so. In future, for my own purposes, I believe I shall generally use Mr. Harza's equation when a close approximation is desired; although it is just as essential to have recourse to Equation 7 as well in order to determine the upper limit of a safe value. If only one equation be used the preference ought to be given to the latter. One may state with considerable precision, and with danger of misleading, that Equation 7 provides for 10 to 20 per cent greater load change than that assumed in the premises. This error may be materially reduced if considered worth while, by the exercise of a little judgment, which may be greatly assisted by reference to Mr. Larner's excellent work.

59 From a practical point of view, another reasonable way of regarding the error encountered in the use of Equation 7, is through an inspection of the algebraic expression for porportionate load change, which is,

$$1 - \frac{h\,V_1}{(h - y_{\max})\,V_{\max}}$$

where h is the working head for the velocity V_1. $V_{\max}$ is an important item for consideration as well as $Y_{\max}$, and for the same assumed range of $V_{\max} - V_1$ and for a value of $Y_{\max}$ only a very small part of h, say, 5 per cent as it should be in a practical case, errors of even 15 per cent in the computed value of $Y_{\max}$ would not ordinarily mean as much as 1 in the figured percentage of load change. It is never necessary to make errors nearly so great as 15 per cent if one has gone into the subject sufficiently to be qualified to use the equation at all. If one does not foresee and bear in mind the obvious fact that a substitution of $V_{\max}$ for V_2 in Equation 7 will give a value of R too small, one may naturally multiply errors unreasonably by persistently increasing $V_{\max}$ for a fixed V'_0, in order to make it agree with Y which is itself already too large; such a course is inexcusable. A tank designed on the lines of the foregoing study would allow a

trifle excess depth which would permit, on extraordinary occasion, a useful increase in the value assumed for V_{max} and incidentally allow Y_{max} to attain its assigned value; this would take care of slightly extraordinary load change, provision for which would be inexpensive as compared with the somewhat unintentional factor of safety furnished.

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BOOKS

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EXCHANGES

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CATALOGUES

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WESTINGHOUSE ELECTRIC AND MANUFACTURING CO., PITTSBURG, PA. Direct Current Generators, Type S and R (Circular No. 1156). Direct Current Motors Type EM (Circular No. 1138). Polyphase Induction Motors (Circular No. 1118). Westinghouse Type S Distributing Transformers (Circular No. 1157).

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

039 Young man wanted to serve as assistant in the mechanical engineering department of a large technical school. Must be familiar with steam power machinery. Position must be filled immediately. Location, New York.

040 Energetic and capable engineer to build and operate plant; to take complete charge of power house, both electric and ice-making and be accountable for costs of production. Must invest one or two thousand dollars or more. Living quarters in conjunction with plant. Service to begin not later than February 1, 1909. Salary not to exceed $100. Location, Colorado.

041 An engineering company in New York wishes to engage young technically-educated engineer as assistant inspector and tester. One with practical as well as theoretical knowledge of isolated plants preferred. Apply by letter, stating qualifications fully.

MEN AVAILABLE

185 Member, with thorough business training, up-to-date factory manager; good executive and organizer; competent in manufacturing medium and light machinery or in purchasing department; fully qualified to fill position of responsibility. Patentee many devices. Willing to go abroad.

186 Member, familiar with gas engine and producer business, in touch with a large number of Eastern prospects, will consider equitable proposal to represent a reliable company or companies building gas engines and gas producers or both.

187 Junior, technical graduate, married, age 32; 12 years experience consisting of shop work, drafting, erecting and operating large steam and gas engines; desires to change

188 Member who has given his attention to the design of special machinery and tools for manufacturing purposes, wishes a position in this line, or as assistant superintendent.

189 Electrical engineer and superintendent of construction, 8 years in charge of design, construction and operation of electrical refrigeration and ice-making plants, responsible charge. Experience covers period of 18 years.

190 Member, Stevens graduate, at present holding responsible position with large concern, desires to make change. 18 years experience, general manufacturing and foundry work, 5 years power and manufacturing plant construction, including design and equipment. Capable of filling position of manager, general superintendent, or chief engineer in charge of construction work, repair and maintenance of plant and equipment. Chicago or New York preferred. Salary $8000 to $10,000 or its equivalent.

191 Junior, experienced business and sales manager. Willing to go abroad.

192 Junior, American, married, 3½ years general design, fabrication and erection, in Mexico, speaks Spanish, German, Russian, and a little French, would like to make the acquaintance of parties having European or Latin-American connections, with view to present or future engagement.

193 Member, at present assistant works manager of one of the large maufacturing establishments of the country, wishes to make a change. Consulting experience, familiar with power plant and steam engine design; served for some time on the teaching staff of an engineering college.

194 Inspector of construction on water works, or mill work: also extensive practical experience in management of large undertakings. and upbuilding of one of the foremost American textile factories. Railroad experience.

195 British manufacture of American patents: member, technical graduate with 12 years varied and practical experience, especially mining machinery and contractor's plants, 5 years in England managing European branch of American house. Now in America; would like to hear from American owners of British patents expecting to start manufacture in England under requirements of new British Patents Act or desirous of placing British rights in the hands of responsible manufacturers upon royalty or other basis.

196 Member, open for engagement, specialty supervision in hydraulic engineering as manager and superintendent. Extensive experience in construction of pumping machinery, designing and equipment of manufacturing plants for economical production.

197 Associate member, 5 years experience as consulting engineer in Mexico, speaks Spanish fluently, desires engagement, preferably South America or Mexico.

198 Sales manager and engineer, member, open for engagement after January 1, 1909. Established in New York; knows manufacturing and selling, competent to lay out and estimate work. Broad experience through travel for special machinery and plant equipments. Manufacturers and bank references.

199 Member, technical graduate, 20 years experience, 10 years in charge of design, construction and maintenance industrial and steam plants, crushing, hoisting, conveying and transmission machinery; reinforced concrete.

200 Junior member, technical graduate in mechanical engineering, married, ten years varied experience, including shop work, drafting and as superintendent of construction of buildings and complete steam and electrical plant, the erection and testing of steam engines. At present employed as Chief Engineer of extensive plant. Desires position in East as assistant manager or superintendent of industrial concern or with consulting engineer.

201 Superintendent or manager experienced in the manufacturing machine shop, gray iron and brass foundries producing large output of good quality.

202 Associate, technical graduate with experience in engineering laboratory, drafting room, miscellaneous work and special machinery design, experimental and testing work, wishes to connect with consulting or contracting engineer in Chicago. Has had charge of work. Can invest small sum if mutually agreeable.

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TAYLOR, Wyatt W. (Associate, 1908), Cons. Engr., 2 Rector St., New York, N. Y

THRALL, Georg C. (1908), Genl. Mgr., Taylor Water Tube Boiler Co., Detroit, Mich.

WAERN, Adolph Wm. (1908), Bethlehem Steel Co., and *for mail*, 210 Broad St., Bethlehem, Pa.

WINTZER, Rudolph (1908), Power and Mining Mchy. Co., 115 Broadway, New York, N. Y.

WORTH, Charles C. (1908), Asst. Supt. Constr., N. Y. Edison Co., 55 Duane St., New York, N. Y.

WUERFEL, Geo. D. (1908), Member of Firm, Wuerfel & Miggett, Cons. and Mech. Engrs., 1125 Ohio Bldg., Toledo, O.

Gas Power Section

BROWN, Ernest C. (Affiliate, 1908). Pres., Rotary Meter Co., Editor, Progressive Age., 280 Broadway, New York, N. Y.

JONES, Hiram K. (Affiliate, 1908) Gas. Eng. Dept., People's Gas Light and Coke Co., 157 Michigan Ave., and 2521 Lowell Ave., Chicago, Ill.

PROMOTIONS

BOWEN, Harrison Seba (1895; 1908), Mech. Engr., R. W. Hunt & Co., 1121 Rookery Bldg., Chicago, and Wilmette, Ill.

EARLE, Samuel Broadus (1905; Associate, 1908), Assoc. Prof. of Mech. Engrg., Clemson College, Clemson College, S. C.

GORDON, Rea, M. (1902; Associate, 1908), Asst. Engr. of Tests, Solvay Process Co., and *for mail*, 533 S. Salina St., Syracuse, N. Y.

HALL, Rodney D. (1902; 1908), Asst. to Ch. Engr., The Snow Steam Pump Co., and *for mail*, 61 Anderson Pl., Buffalo, N. Y.

PERRY, Frank B. (1902; 1908), Mill Power Dept., Genl. Elec. Co., 84 State St., Boston, and *for mail*, 71 Athol St., Allston, Mass.

RICHMOND, Knight C. (1891; 1908), Cons. Engr., 526 Banigan Bldg., and 204 Angell St., Providence, R. I.

VAN WINKLE, Edward (1904; 1908), Cons. Patent Engr., Flat Iron Bldg., Madison Square, New York, and The Stuyvesant, 483 Park Pl., Brooklyn, N. Y.

WILKINS, I. Chester G. (1899; 1908), Mech. Engr., Thompson-Starrett Co., 51 Wall St., New York, N. Y.

RESIGNATIONS

SCHAEFFER, Louis C.

DEATHS

JONES, Edwin H.

COMING MEETINGS

AERONAUTIC SOCIETY

January 13, 20, 27, February 3, evenings, office Dr. Greene, 39th St. and Fifth Ave., New York. Secy., Wilbur R. Kimball.

AMERICAN ASSOCIATION FOR THE ADVANCEMENT OF SCIENCE

December 28, 1908–January 2, 1909, annual meeting, Baltimore, Md. Secy., L. O. Howard, Smithsonian Institution, Washington, D. C.

AMERICAN GAS POWER SOCIETY

January 26, quarterly meeting, Minneapolis, Minn. Secy., R. P. Gillette.

AMERICAN GEOGRAPHICAL SOCIETY

January 26, 29 W. 39th St., New York, 8 p.m. Acting Secy., Geo. C. Hurlbut, 15 W. 81st St.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

January 15, monthly meeting, Toronto Section. Secy. *pro tem.*, W. H. Eisenheis, 1207 Traders' Bank Bldg.

AMERICAN INSTITUTE OF MINING ENGINEERS

February, 29 W. 39th St., New York.

AMERICAN MATHEMATICAL SOCIETY

January 1, 2, Chicago Section, University of Chicago, Chicago, Ill. General Secy., F. N. Cole.

AMERICAN SOCIETY OF CIVIL ENGINEERS

January 20, February 3, 220 W. 57th St. Secy., C. W. Hunt.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

January 19–21, annual meeting, 29 W. 39th St., New York. Secy., W. M. Mackay.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

January 12, February 9, monthly meetings, 29 W. 39th St., New York, 8 p.m. Secy., Calvin W. Rice.

ASSOCIATION OF AMERICAN GEOGRAPHERS

January 1–2, annual meeting, Baltimore. Secy., Albert Perry, Brigham Hamilton, N. Y.

BOSTON SOCIETY OF CIVIL ENGINEERS

January 27, monthly meeting, Tremont Temple. Secy., S. E. Tinkham, 60 City Hall.

CANADIAN RAILWAY CLUB

February 2, Montreal, Que. Secy., Jas. Powell, Care Grand Trunk Ry.

CANADIAN SOCIETY OF CIVIL ENGINEERS

January 14, General Section Meeting, 413 Dorchester St. W., Montreal, Que. Secy., Prof. C. H. McLeod.

CANADIAN SOCIETY OF CIVIL ENGINEERS, Manitoba Branch

January 15, February 5, semi-monthly meetings, University of Manitoba. Secy., E. Brydone Jack.

CANADIAN SOCIETY OF CIVIL ENGINEERS, Quebec Branch
January 15, annual meeting. Secy., P. E. Parent, P. O. Box 135, Quebec.

CANADIAN SOCIETY OF CIVIL ENGINEERS, Toronto Branch
January 7, annual meeting, January 28, regular meeting, 96 King St. W. Secy., T. C. Irving, Jr.

CAR FOREMEN'S ASSOCIATION OF CHICAGO
January 11, February 8. Secy., Aaron Kline, 326 N. 50th St.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS
February 1, Indianapolis, Ind. Secy., G. B. Staats, Care Penna. Lines.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS
January 11, February 8, Kansas City, Mo. Secy., F. H. Ashley, Gumbel Bldg.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS
January 14, Toledo, O. Secy., H. M. Ellert.

CENTRAL RAILWAY AND ENGINEERING CLUB OF CANADA
January 19, Rossin House, Toronto, Ont. Secy., C. L. Worth, Room 409, Union Sta.

CLEVELAND ENGINEERING SOCIETY
January 12, February 9, monthly meetings, Caxton Building. Secy., Joe C. Beardsley.

COLORADO SCIENTIFIC SOCIETY
February 6, monthly meeting, Denver. Secy., Dr. W. A. Johnston, 801 Symes Bldg.

ENGINEERING ASSOCIATION OF THE SOUTH
January 19, monthly meeting, Nashville Section, Carnegie Library Bldg. Section Secy., H. H. Trabue, Berry Blk., Nashville.

ENGINEERING SOCIETY OF THE STATE UNIVERSITY OF IOWA
February 2, monthly meeting, Iowa City, Ia. Secy., Dean Wm. G. Raymond.

ENGINEERS' CLUB OF BALTIMORE
February 6, monthly meeting. Secy., R. K. Compton, City Hall.

ENGINEERS' CLUB OF CENTRAL PENNSYLVANIA
February 2, monthly meeting, Gilbert Bldg., Harrisburg. Secy., E. R. Dasher.

ENGINEERS' CLUB OF CINCINNATI
January 21, 25 E. 8th St. Secy., E. A. Gast, P. O. Box, 333.

ENGINEERS' CLUB OF TORONTO
January 14, etc., weekly meetings, 96 King St. W., Toronto, Ont. Secy., R. B. Wolsey.

ENGINEERS' CLUB OF PHILADELPHIA
January 23, February 6, 1317 Spruce St. Secy., H. G. Perring.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA
January 19, annual meeting, 803 Fulton Bldg., Pittsburgh, Pa. Secy., E. K. Hiles.

ILLINOIS SOCIETY OF ENGINEERS AND SURVEYORS
January 27–29, annual meeting, Great Northern Hotel, Chicago. Secy., E. E. R. Tratman, 1636 Monadnock Blk.

ILLUMINATING ENGINEERING SOCIETY
January 14, monthly meeting, New York Section, 29 W. 39th St., 8 p.m. Secy., P. S. Millar.

INDIANA ENGINEERING SOCIETY
January 14–16, Commercial Club, Indianapolis. Secy., Chas. Brossmann, 43 Union Trust Bldg.

INTERNATIONAL WATER LINES PASSENGER ASSOCIATION
January, Niagara Falls, Ont. Secy., M. R. Nelson, 379 Broadway, New York.

INTERSTATE SHIPMASTERS' ASSOCIATION
January, Washington, D. C.

IOWA ENGINEERING SOCIETY
January 13–14, annual meeting, Waterloo. Secy., A. H. Ford, Iowa City.

LOUISIANA ENGINEERING SOCIETY
January 9, annual meeting, 323 Hibernia Bldg., New Orleans. Secy., L. C. Datz.

MASSACHUSETTS STREET RAILWAY ASSOCIATION
January 13, February 10, Boston. Secy., Charles S. Clark, 70 Kilby St.

MICHIGAN ENGINEERING SOCIETY
January 12-15, Ann Arbor, Mich. Secy., Alba L. Holmes, Grand Rapids, Mich.

MUNICIPAL ENGINEERS OF THE CITY OF NEW YORK
January 27, 29 W. 39th St., 8:15 p.m. Review of the Work of the Various Departments of the City Government for 1908, Robt. Ridgway. Secy., C. D. Pollock.

NATIONAL ASSOCIATION OF AUTOMOBILE MANUFACTURERS
January 20, New York. Secy., C. C. Hildebrand, 7 E. 42d St.

NATIONAL ASSOCIATION OF CEMENT USERS
January 11–16, annual convention, Cleveland, O. Secy., George C. Wright, Harrison Bldg., Philadelphia, Pa.

NEW ENGLAND RAILROAD CLUB
January 12, February 9, Boston, Mass. Secy., Geo. H. Frazier, 10 Oliver St. Paper: Single Phase Railway System, N. W. Storer.

NEW ENGLAND STATES GAS ENGINEERS' ASSOCIATION
February 17, Boston, Mass. Secy., M. Gifford, 26 Central Sq., E. Boston.

NEW ENGLAND STREET RAILWAY CLUB
January 28, monthly meeting, American House, Boston, Mass.; March 25, annual meeting. Secy., John J. Lane, 12 Pearl St.

NEW ENGLAND WATERWORKS ASSOCIATION
January 13, February 10, regular meetings. Secy., Willard Kent, Tremont Temple, Boston, Mass.

NEW YORK RAILROAD CLUB.
January 8, 15, 29 W. 39th St. Paper: Education and Organization of Railway Engineering Labor. Secy., H. D. Vought, 95 Liberty St.

NEW YORK SOCIETY OF ACCOUNTANTS AND BOOKKEEPERS
January 12, etc., weekly meetings, 29 W. 39th St., 8 p.m. Secy., T. L. Woolhouse.

NORTHERN RAILWAY CLUB
January 23, Commercial Club Rooms, Duluth, Minn. Secy., C. L. Kennedy.

NORTHWESTERN ELECTRICAL ASSOCIATION
January 15–16, Hotel Pfister, Milwaukee, Wis. Papers: Series Tungsten Lighting; Increase of Central Station Load by Sub-Station to Nearby Hamlets; Keeping Track of Supplies and Contracting; The Tungsten Lamp;

Commercial Development of Practical Knowledge for Practical Men; Storage Battery Auxiliaries for Small Central Stations. Secy. Roger Kimball.

NORTHWEST RAILWAY CLUB

January 10, St. Paul, February 9, Minneapolis, Minn. Secy., T. W. Flannagan, Care Soo Line, Minneapolis.

NOVA SCOTIA SOCIETY OF ENGINEERS

January 14, monthly meeting, Halifax. Secy., S. Fenn.

OHIO ENGINEERING SOCIETY

January 26–28, Board of Trade Bldg., Columbus. Secy., Paul Hansen, 912 Harrison Bldg.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS

January 26, monthly meeting, 48 Snow St.; June 22, annual meeting. Secy., T. M. Phetteplace.

PURDUE MECHANICAL ENGINEERING SOCIETY

January 20, February 3, fortnightly meetings, Purdue University, Lafayette, Ind., 6:30 p.m. Secy., L. B. Miller.

RAILWAY CLUB OF PITTSBURGH

January 22, monthly meeting, Monongahela House, Pittsburgh, Pa., 8 p.m. Secy., J. D. Conway, Genl. Office, P. & L. E. R. R.

RENSSELAER SOCIETY OF ENGINEERS

January 15, 29, 257 Broadway, Troy, N. Y. Secy., R. S. Furber.

RICHMOND RAILROAD CLUB

January 11, February 8, Richmond, Va. Secy., F. O. Robinson, Care C. & O. Ry.

ST. LOUIS RAILWAY ASSOCIATION

January 8, Palace of Transportation. Paper: The Ethics of Railroading, E. F. Kearney. Secy., B W. Frauenthal.

SCRANTON ENGINEERS' CLUB

January 21, Board of Trade Bldg. Secy., A. B. Dunning.

SHORT LINE RAILROAD ASSOCIATION

February 1, New York. Secy., Cromwell G. Macy, Jr., Nantucket Central R. R. 257 Broadway.

SOCIETY OF AUTOMOBILE ENGINEERS

Week of January 14, annual meeting, New York. Secy., C. B. Hayward, 915 Flatiron Bldg.

SOUTHERN AND SOUTHWESTERN RAILWAY CLUB

January 21, Atlanta, Ga. Secy., A. J. Merrill, 218 Prudential Bldg.

TECHNICAL SOCIETY OF BROOKLYN

January 15, February 5, bi-monthly meetings, Arion Hall, Arion Pl., Brooklyn, N. Y., 8:30 p.m. Paper: Electrical Power Transmission, Henry Pikler. Pres., M. C. Budell

TECHNICAL SOCIETY OF THE PACIFIC COAST

February 5, semi-monthly meeting, San Francisco, Cal. Secy., Otto von Geldern, 1978 Broadway.

TECHNOLOGY CLUB OF SYRACUSE

January 12, February 9, monthly meeting, 502 Bastable Blk., 8 p.m. Secy., Robert L. Allen.

WESTERN RAILROAD ASSOCIATION

January, annual meeting. Secy., Edw. P. Amory, 1330 Marquette Bldg., Chicago, Ill.

WESTERN RAILWAY CLUB

January 19, monthly meeting, Auditorium Hotel, Chicago, Ill., 8 p.m. Secy., Jos. W. Taylor, 390 Old Colony Bldg.

WESTERN SOCIETY OF ENGINEERS

January 13, etc., bi-weekly meetings, 1737 Monadnock Blk., Chicago. Secy., J. H. Warder.